HYBRIDIZED ALL THE YEAR-ROUND AIR CONDITIONING SYSTEM

by

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CERTIFICATE

Certified that this work on 'HYBRIDIZED ALL THE YEAR-ROUND AIRCONDITIONING SYSTEM' by Lalit Kumar Pothal has been carried out under my supervision and that this has not been submitted elsewhere for a degree.

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- Lalit Kumar Pothal

CONTENTS

PAGE

PAGE

LIST OF TABLES

LIST OF FIGURES

PAGE

NOMENCLATURE

A	Frontal area m^2
A_d	Area of duct
COP	Coefficient of performance
h	Enthalpy kJ/kg
$\dot{m}$	Mass flow of refrigerant kg/sec.
$_d$	Desert cooler efficiency
P	Compressor work kW
PI	Performance index
$\dot{Q}_c$	Refrigeration effect kJ/min.
$\dot{Q}_h$	Heat rejection kJ/min.
S	Entropy kJ/kg.K
T_{db}	Dry bulb temperature of air, C
T_{wb}	Wet bulb temperature of air, C
U	Overall heat transfer coefficient kcal/h m^2.C
V	Velocity of discharge air m/min.
ν	Specific volume of refrigerant kg/m^3
$_c$	Compressor efficiency

ABSTRACT

A hybridized all the year-round airconditioning system was fabricated, commissioned and tested for operation in the heat pump and airconditioning modes. For the heat pump operation it took 1480, 1600, 1540 and 1850 W when refrigerants used were R-12, R-22, a mixture of R-12 and R-22 in the ratio of 41% : 59% and 51% : 49% by weight, respectively. COPs were found to be 2.12, 2.48, 2.04, 1.68 and corresponding PIs were 3.12, 3.54,2.97 and 2.79, respectively. In the airconditioning mode of operation, it took 1650 W with the mixture of refrigerants R-12 and R-22 in the ratio of 51% : 49% by weight. COP and PI were found to be 1.48 and 2.52 and in the reverse mode operation, it gave the respective values as 1.68 and 2.79.

Economic analyses for energy and cost were carried out for 5, 10 and 100 ton capacity all the year-round airconditioning systems. Saving in energy as well as its running cost of the order of 62.66% supports the development of this type of system in place of conventional airconditioning plant.

However the present system incorporates valves, dampers, water tank, pump, etc. requiring an additional investment to the tune of Rs. 2400 for 1.5 ton unit. But this additional investment is compensated by the saving in electric bill within 2 years.

CHAPTER-1

INTRODUCTION

1.1 NEED FOR AIRCONDITIONING

From time immemorial, man has been striving to better his life-style. Over the ages with the rapid strides of Science and Technology the frantic endeavour for controlling his immediate environment has culminated in bringing out the devices for Heating, Refrigerating and Airconditioning. These are no longer luxuries rather obvious necessities of the modern era. In general, air-conditioning is defined as the simultaneous control of temperature, humidity, cleanliness and air motion. Mechanical cooling, ventilation, or heating must also be provided to render human comfort to prevent deterioration of machinery components, to prevent excessive pressure built-up in containers, to stop contamination from leaking out of a confined space and to maintain specified temperatures for electrical and electronic equipment. Thus, airconditioning can be classified as comfort and industrial. Undoubtedly the need for airconditioning has proved to be one of the greatest exigencies of the human-being in the light of the increased demand for productivity, superior quality products and comfortable environment.

1.2 COMFORT REQUIREMENT AND ENERGY CONSTRAINT

The requirement of comfort airconditioning is increasing extensively due to emphasis on the better quality of life and productivity. On the other hand the cost for airconditioning is becoming a limiting factor due to energy crises. The engineers and scientists are very much concerned and appalled at the sight of fast exhausting sources of energy. Thus a compromise between energy requirement and comfort level has been made [15]. The inside design conditions T_{db} = 30 C and ϕ = 60% for comfort have been suggested for an average activity of man [4,16]. The higher value than the existing data has been supported by Whitner [4] and ASHARE comfort chart values. Of course the air velocity has been enhanced to 0.7 m/s or higher as against the near static air velocity of 0.13 m/s used presently for comfort airconditioning.

In India Malhotra [2] has obtained the effective temperature for hot and humid as well as hot and dy climate based on recording of 75% votes. These values are presented in table 1.1.

Table 1.1

Comfort conditions for hot and humid as well as hot and dry climate

Sl.No.	Level of comfort	Effective temperature ET in C	
		hot & humid	hot & dry
1	Warm & unpleasant	27.0	26.7 - 28.3
2	Comfortable & pleasant (upper level)	24.5 - 25.0	24.4 - 26.6
3	Comfortable & pleasant (lower level)	22.0 - 22.5	21.1 - 24.3

Extensive work has been done by Tanabe and Kimura [10]. Their comfort conditions for the hot-humid conditions are given in Table 1.2. They kept energy conservation in mind while recommending these values.

Table 1.2

S.No.	Experimental conditions	Air motion m/s
1	27 C, ϕ = 50%	0.5
2	29 C, ϕ = 50%	1.2
3	31 C, ϕ = 50%	1.6

1.3 REVIEW OF PREVIOUS WORK

Human beings can only work and live with comfort and efficiency within a restricted set of physical conditions. Unless these conditions are met, innumerable problems do crop up causing fatal to human race.

Statistical studies and others have substantiated the need for the airconditioning. A study conducted by a team of psychologists and environmentalists suggests that there were an improvement in overall learning phenomennon and working ability [1,16]. The air-conditioning can be divided into the summer airconditioning and

winter airconditioning. The conventional summer airconditioning uses a refrigeration system and a dehumidifier against a heat pump and a humidifier used for winter. By and large, the term airconditioning in Indian context chiefly refers to the control of temperature in a down-ward direction i.e., when the hot air in summer can be cooled and dehumidified. Thus the concept of summer airconditioning is most prevalent in our country.

Comfort airconditioning for India and the process requirements for different climatic zones reveal that the most populous and importan climatic region is monsoon type with dry winters [6,7]. Generally, the present practice in the region is to either use a window air-conditioner from April to October or a desert cooler from April to June to achieve comfort conditions. During winter the normal practice is to use electrical heaters which are most convenient method to serve the purpose. But this causes the misuse of precious electrical energy at the cost of some others prior sectors.

1.4 SCOPE OF THE PRESENT WORK

CONCEPT OF ALL THE YEAR-ROUND AIRCONDITIONER

As a matter of fact, the extensive work on airconditioning in the light of energy conservation has been conducted by Prasad [5,7], Ramamoorthi [15], Misra [18], Khandelwal [7], Pandey [9]. These studies became possible due to acceptance of high comfort conditions inside room.

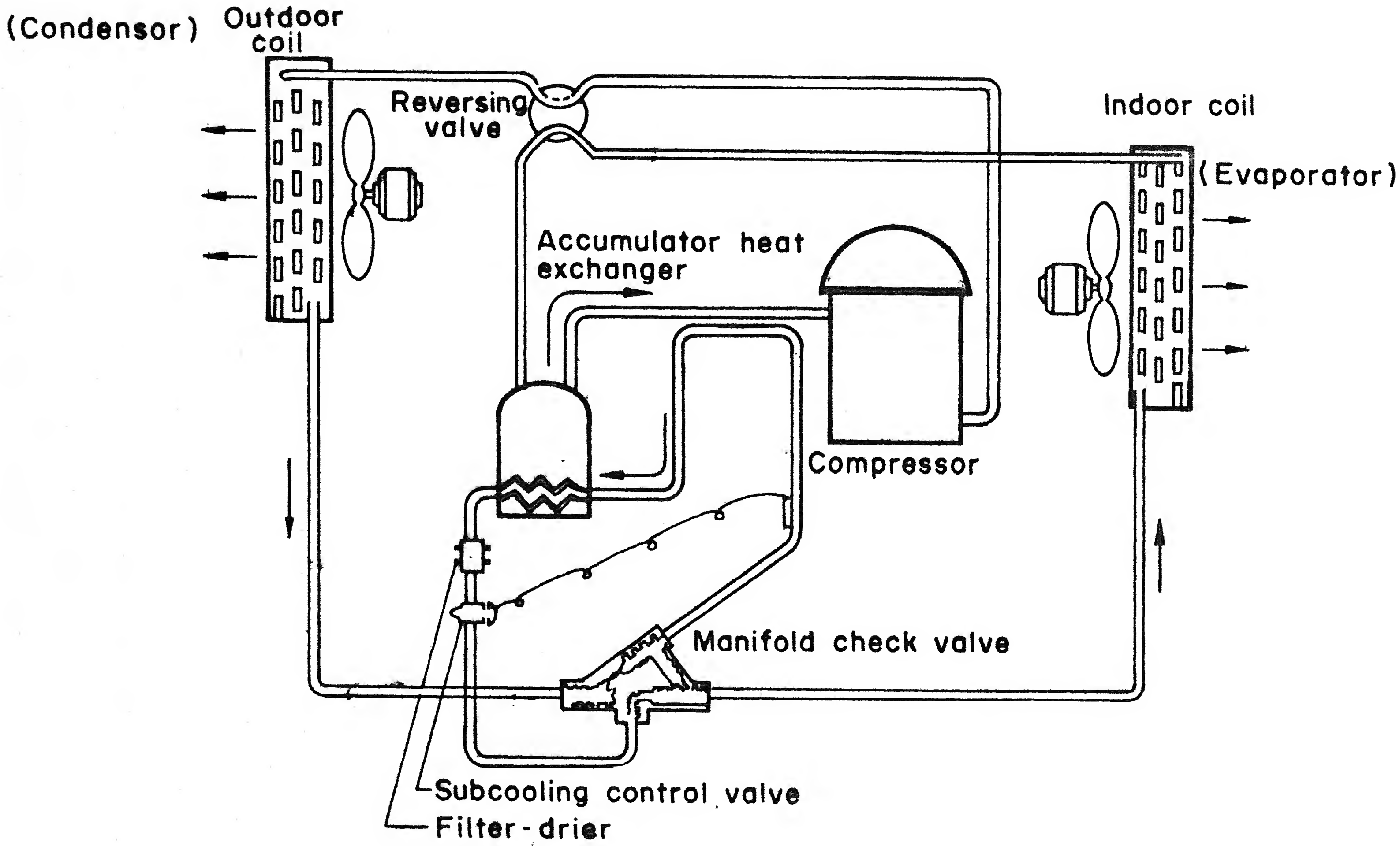

Fig 1.1(a) Heat pump system operating as a cooling system.

(a) One can reach the comfort zone with the help of a desert cooler from April to June and October.

(b) From July to September, the evaporative cooling process is invariably ineffective. The comfort zone can only be approached by the conventional cooling and dehumidifying system.

(c) For the winter months i.e., from November to February, one needs heating to feel comfortable. The heating can be effected by the following methods:

(i) Electrical heating,
(ii) Fuel fired heating,
(iii) Solar heating,
(iv) Steam heating,
(v) Heat pump for heating, etc.

The electrical heating should be discouraged as far as possible as it provides only 30% of the heating per unit of fuel burnt. On the otherhand, the heating by burning fuel is desirable but due to smokes and other inconveniences in its use, does not gain the popularity. The solar heating is still expensive and available for much less than half a day only. Thus the final choice rests with the fifth factor. It is reversing the airconditioning cycle as shown in figure 1.1(b). If the COP of the refrigeration cycle is 2, the performance index of the reverse cycle equals 3(= COP + 1).

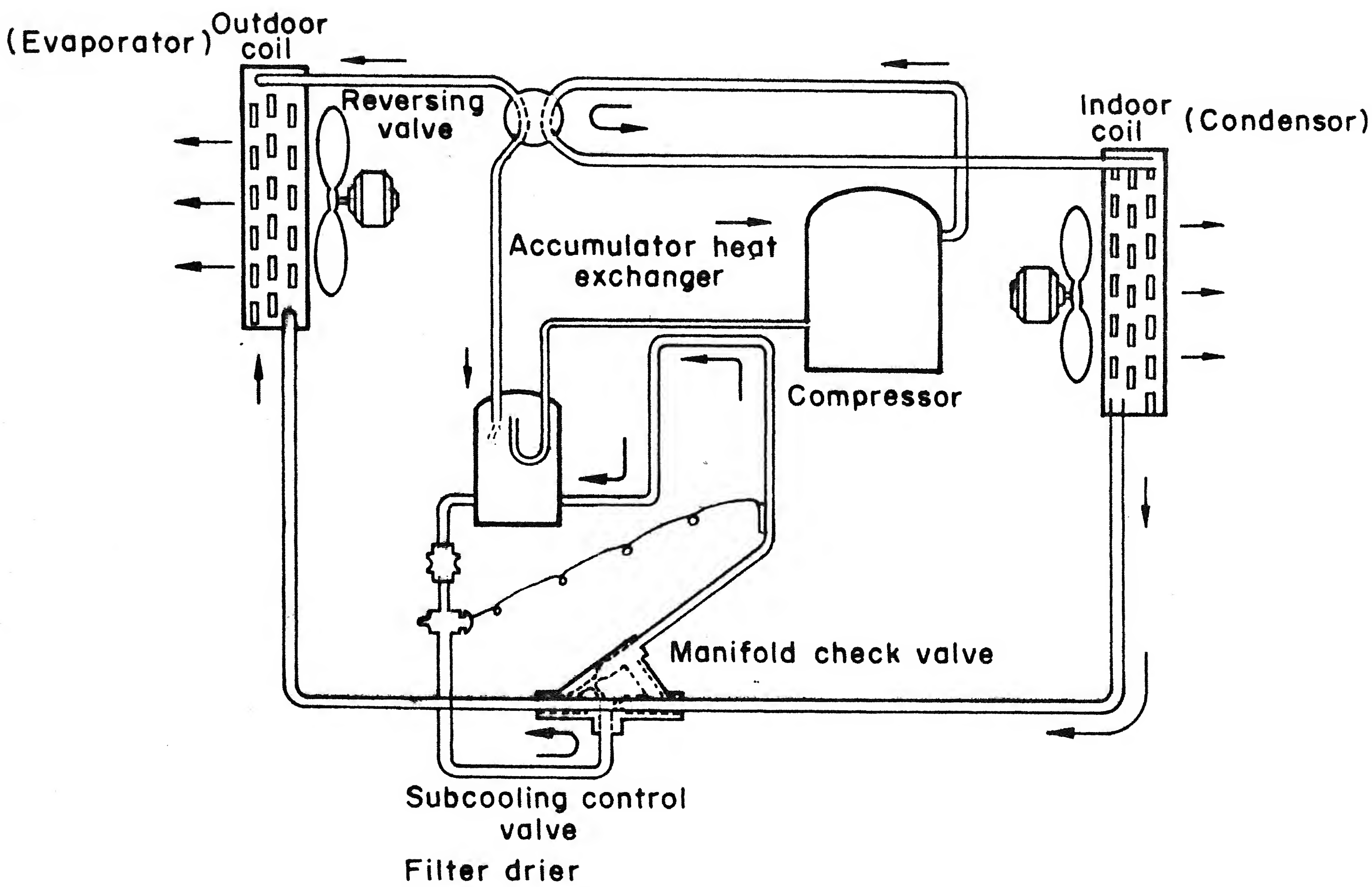

Fig1.1(b) Schematic representation of a heating cycle

Hence for each kW of electric energy one can get 3 kW of heating by the conventional airconditioner.

Based on the ambient and inside conditions for places like Kanpur it has been found that an evaporative cooler renders comfort conditions requiring much less energy than that of the conventional airconditioner [6,9]. The use of the conventional airconditioner is not the correct use for the place where the climate is hot and dry. However, in the rainy season the desert or evaporative cooler becomes ineffective as it needs cooling and dehumidification. Thus the conventional airconditioner comes into use.

The above findings are in perfect conformity with many investigators such as Malhotra [2], Ramamoorthi [15], Misra [17], Khandelwal [7], Prasad [1] and Tanabe & etal [10]. With above said considerations,the present attempt has been in the direction of developing a hybridized all the year round airconditioning system or simply three-in-one i.e., cooling for the hot-dry, cooling for hot-humid and heating for winter.

CHAPTER-2

BASIC PROCESSES AND SYSTEMS FOR THEIR ACHIEVEMENTS

2.1 DESCRIPTION

From energy conservation point of view the following processes are to be accomplished to achieve the desired inside condition:

(i) Humidification process by evaporative cooling for hot-dry season

(ii) Cooling and dehumidification for hot-humid and

(iii) Heating and humidification for winter season.

Because these processes are the real requirements for the major parts of India as can be seen from the meterological conditions [9]. The processes are described in detail in the subsequent sections.

2.1.1 EVAPORATIVE COOLING

The adiabatic evaporation is the well known process for rendering cool air in the hot-dry environment. Hence it provides comfort conditions having humidity in air within tolerable limits. Figure 2.1(a) exhibits schematically an evaporative cooling process.

in Fig.2.1(b), the air at state A gets cooled to state B due to evaporation of water. Then heat addition and humidity in the room raises the air state to design condition state point C.

Saturation efficiency or cooling efficiency is the measure of the extent to which leaving air temperature approaches thermo-dynamic wet-bulb temperature of entering air.

$$n_d = \frac{T_{db} - T_{dbg}}{T_{db} - T_{wb}}$$

Where,

n_d = Desert cooler efficiency

T_{db} = Dry-bulb temperature of the entering air

T_{dbg} = Dry-bulb temperature of air leaving the evaporative cooler

T_{wb} = Thermo-dynamic wet-bulb temperature of entering air

2.1.2 MECHANICAL COOLING

The vapour compression cycle is the most widely used refrigeration cycle for airconditioning. In airconditioning, the refrigerant in the liquid state is fed to the evaporator through a throttling device and in the process of changing its state from liquid to vapour, it absorbs and removes heat from the space to be

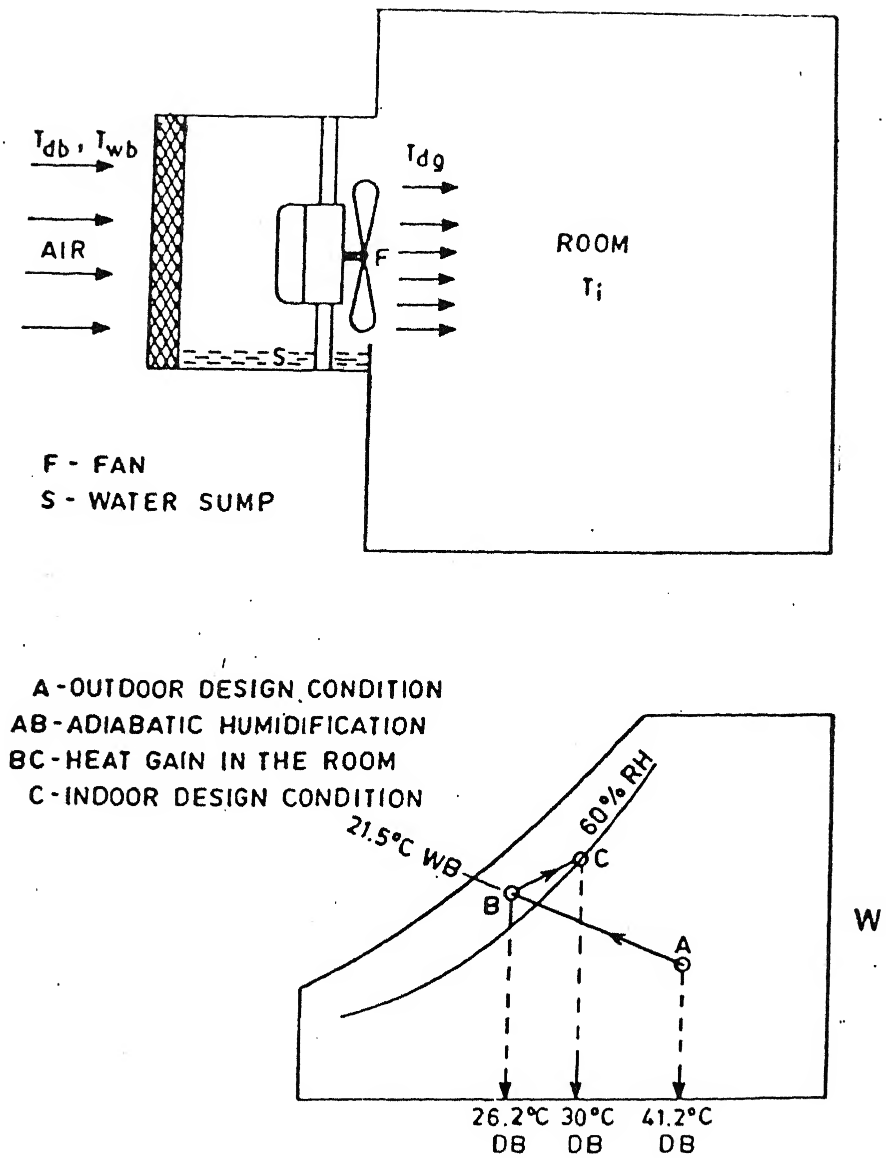

Fig. 2.1. (a) Evaporative cooling arrangement in room.
(b) Evaporative cooling process. [6]

airconditioned. This is also called mechanical refrigeration which is well known vapour compression system. Figure 2.2 shows the system as well as cycle schematically.

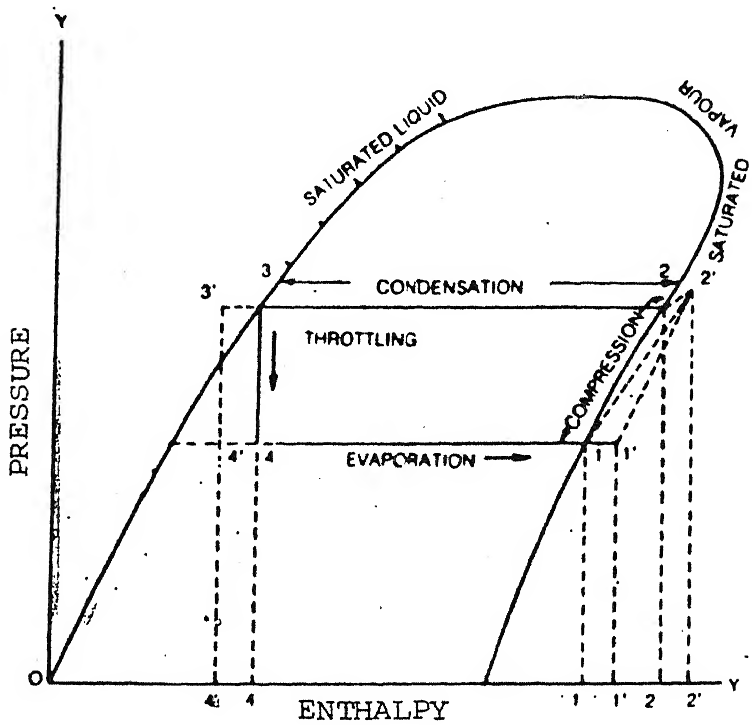

Fig. 2.2. P-h diagram for vapour compression cycle.

Following can be found from the diagram, assuming ideal conditions:

Refrigeration effect = $\dot{m}\ (h_1 - h_4)$

where, $\dot{m}$ = mass flow of refrigerant

h_1 , h_4 = enthalpy of refrigerant at respective state point.

Heat equivalent of work

done in compression $= \dot{m}\ (h_2' - h_1)$ (2-2)

Heat rejected in condensation $= \dot{m}\ (h_2' - h_3)$ (2-3)

Coefficient of performance $= \dfrac{h_1 - h_4}{h_2' - h_1}$ (2-4)

For actual performance the compressor efficiency is calculated from the expression:

$$n_c = \frac{h_{2s} - h_1}{h_2' - h_1} \tag{2-5}$$

Where h_{2s} = Isentropic enthalpy at point 2,

Thus the actual power is given by

$$P = \dot{m}\ (h_2' - h_1)/\ n_c$$

For the more realistic approach, even pressure drops in evaporator, condenser and compressor are also considered. These values are take on the basis of available data in the standard text books [1,8].

2.2 HEAT PUMP CYCLE FOR WINTER

The year round airconditioning comprises (i) Conventional airconditioning and (ii) Heat pump for the winter season.

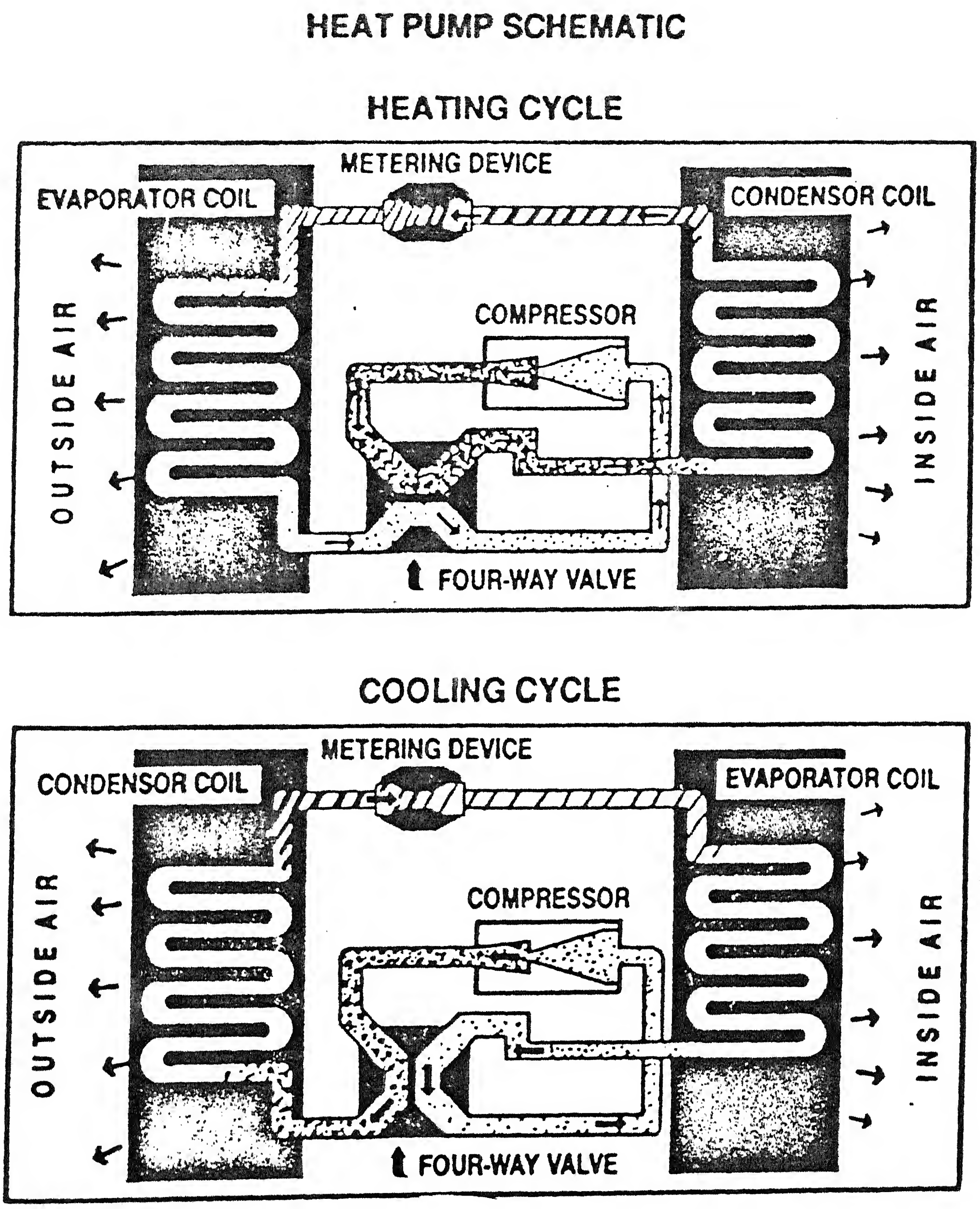

Fig. 2.3(a). Schematic of heating cycle and cooling cycle.
(HVAC Basics)

1. CONVENTIONAL AIRCONDITIONER:

In this mode the evaporator of a standard airconditioner removes heat from the supply air to the conditioned space during the cooling season and dispels it through condenser to outdoor air.

2. HEAT PUMP FOR WINTER:

In this operation the cycle is reversed. The evaporator removes heat from outdoor air, water, or preferably from high temperature source. The heat is then pumped to the condenser which provides heat to the air supply and to the conditioned space.

These two modes of operations are exhibited in Fig. 2.3 for the winter heating and summer cooling. The system incorporates transfer valves to change the operations as per requirements i.e. the winter heating or summer cooling. The transfer valve is a kind of four-way valve which operates by pressure. A soleniod valve controls the pressure at top portion of reversing valve. When solenoid pilot valve is energized, the compressor low-side pressure pushes on the transfer valve four way valve. All three internal valves lifted, producing desired cycle.

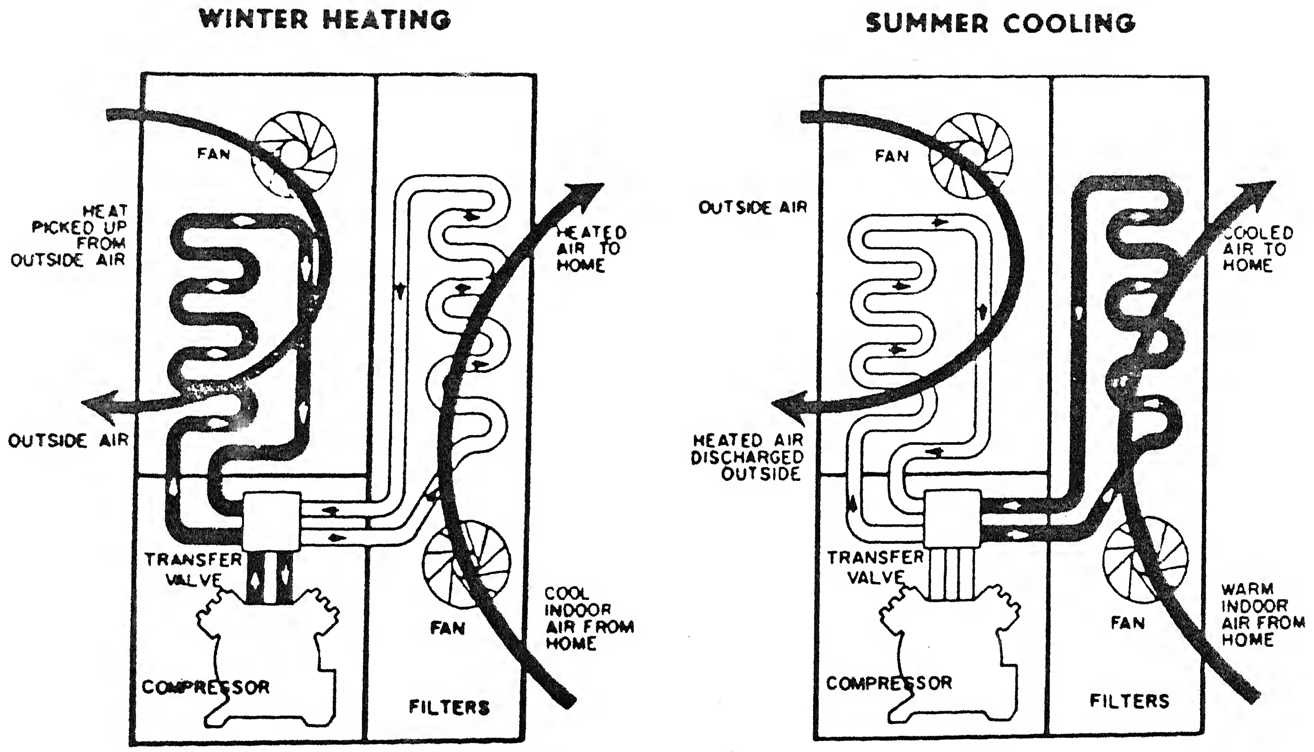

Fig. 2.3.b. Schematic representation of heat pump and airconditioning modes of operation. [11]

Several types of comfort cooling systems are in use. We have chosen the window units since it is very popular comfort cooler for a small capacity. The system consists principally of:

1. An air cooled condensing unit
2. Evaporator
3. Filters
4. Refrigerant
5. Piping and
6. Controls

The condensere is located in the section of the cabinet that is outside the building. The outside air is forced over the condense by a fan. Inside the room another fan draws in air through a filter and forces it over the evaporator coil for cooling and dehumification of air. The condenser and evaporator fans are generally driven by the same electric motor. Fig. 2.4 illustrates a modern window airconditioner.

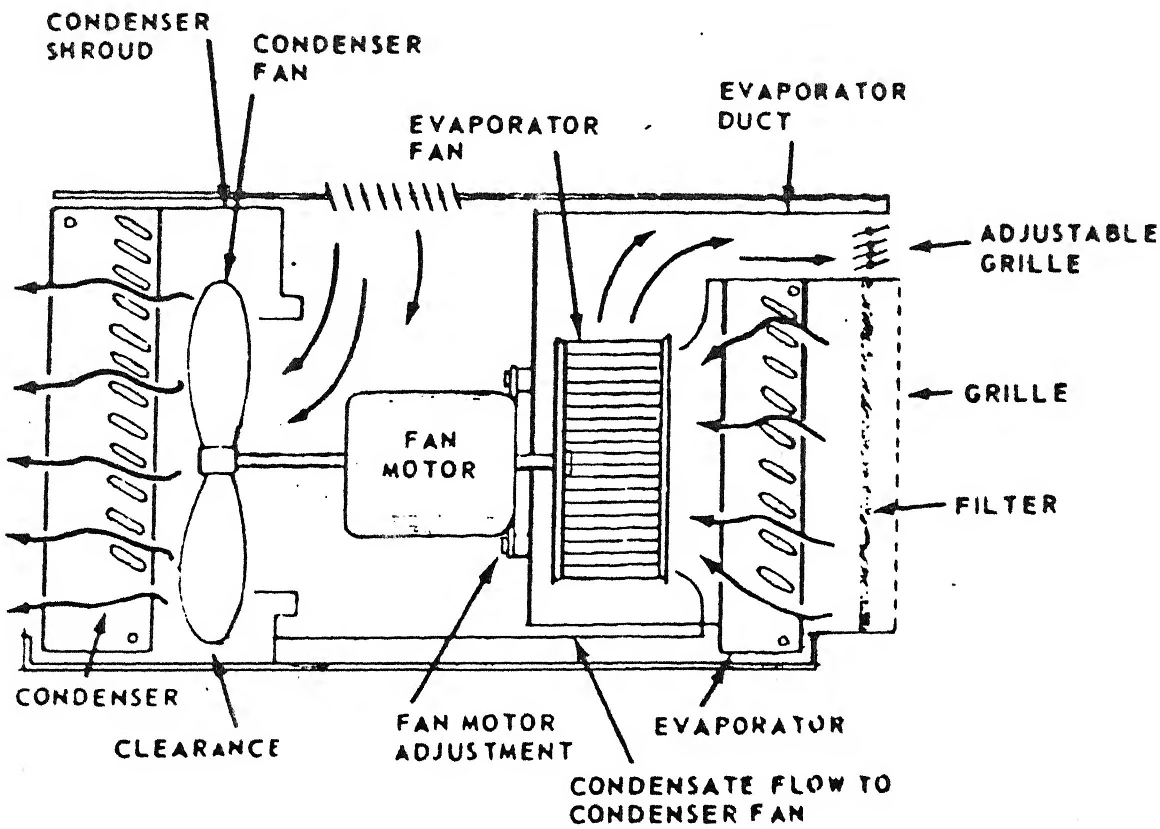

Fig. 2.4 Schematic drawing of an airconditioner. [11]

CHAPTER-3

SYSTEM DESIGN AND EXPERIMENTAL SET-UP

3.1 INTRODUCTION

Technologically superior components cannot perform optimally, unless it is meticulously matched to suit proper application and installation. This requires considerable informations viz.materials, components, design steps, optimum combinations besides thermodynamic feasibility, economic feasibility, physical and social feasibility. Thus, for a system to work as an airconditioner or heat pump, the condenser size and design, the evaporator size and design, the system piping which includes the capillary size, all have to work in a combined manner to produce the designed cooling/heating or tonnage.

The present system comprises the conventional airconditioner, evaporative cooler, piping and manual shut-off valves for reversal cycle operation. The various components are as follows:

- (i) Compressor
- (ii) Evaporator
- (iii) Expansion device (double capillary tube)
- (iv) Condenser
- (v) Blower and fan

(vi) Water circulation sub-system

(vii) Valves fan numbers of Ø 12.7 mm and two Ø 6.3 mm

(viii) Copper pipes

(ix) Ducting (Aluminium)

(x) Instruments and control

3.2 DESIGN/SELECTION OF SYSTEM COMPONENTS

Figure 3.1 shows the schematic diagram of the prototype of 1.5 ton complete airconditioner. For standardization and easy availability of the components in the market, mostly standard equipment have been used. Selection of components are as under.

3.2.1 SELECTION OF COMPRESSOR

The compressor is known as the heart of the refrigeration system. The role played by a compressor cannot be based on its individual performance, but has to match with other components during operation to suit the compressor output. To make the compressor highly efficient, by producing maximum output with minimum power consumption, having durability and better serviceability, making it light weight and compact, has been the endeavour of all designers.

The purpose of compressor is to maintain a desired evaporator pressure corresponding to the requirement of low temperature.

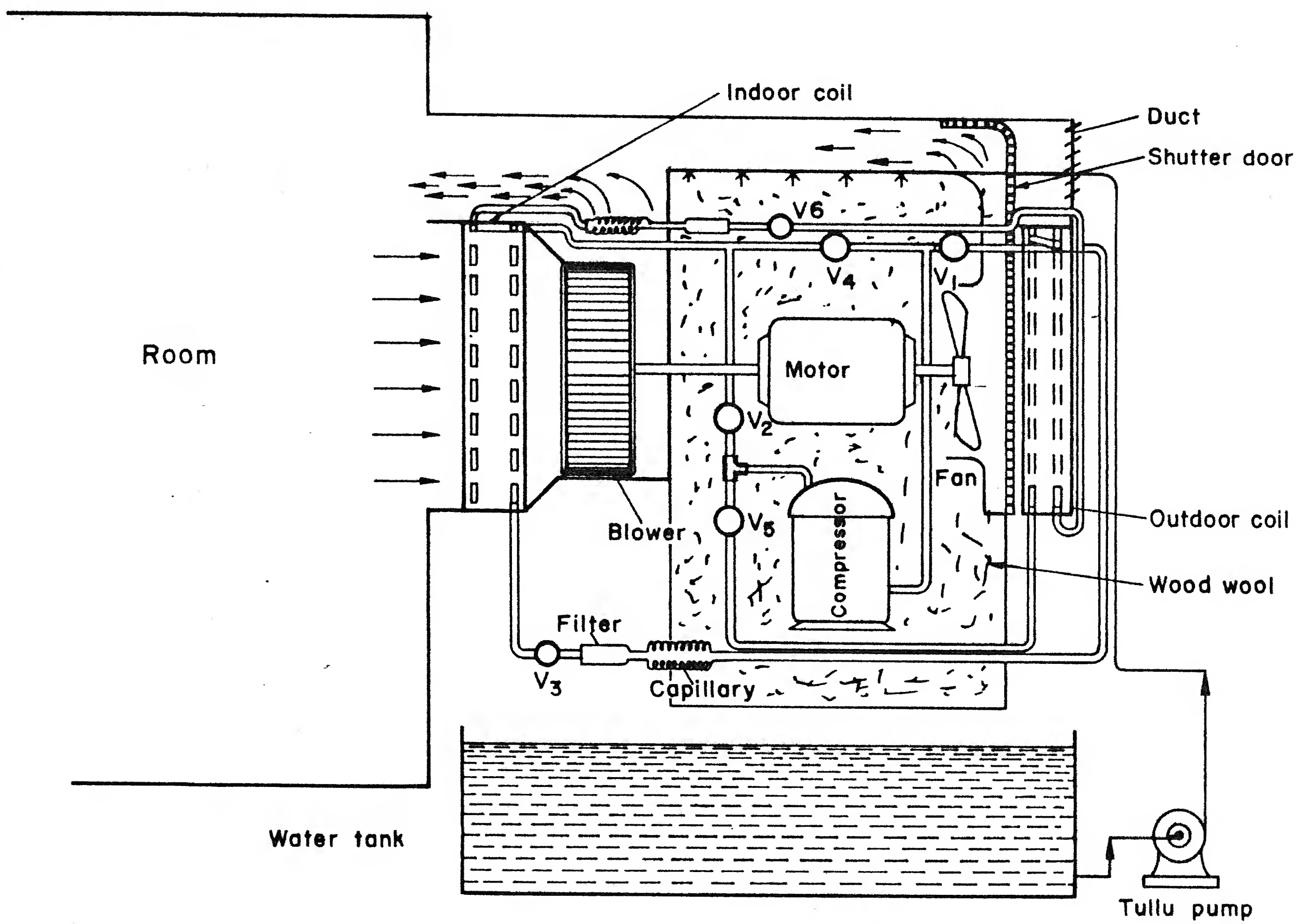

Fig. 3.1 Schematic diagram of hybridized all the year round air conditioning system.

In the present set-up the hermetic compressor is used with following specifications.

MODEL NE 1800 BB (1.5 T)

Nominal Performance	Airconditioning	Heat Pump
Input power	1800	1470
Input current (A)	9	7.5
Rating conditions		
Evaporating Temperature	7.2 C	-1 C
Condensing Temperature	55 C	44 C
Mechanical Data		
Displacement volume per revolution	34.5 CC	
RPM	2925	
Bore x stroke	3.45 cm x 1.86 cm	
Application		
Evaporating temperature range	3.9 C to 12 C	

3.2.2 EVAPORATOR SELECTION

It is required of an evaporator of small size to transfer enough heat. The refrigerant should not leave the evaporator as

a liquid in order that dry compression is effected. The size and arrangement of the pipe should be adjusted as to cause easy oil return to the compression crank case. The pressure loss should be as low as possible, corrosion and fouling of the inside and outside surfaces should be minimum.

In the present set-up a 1.5 ton cooling capacity fin and tube type evaporator has been used. Specifications are as under:

Number of tubes	=	20
Tube outside diameter	=	1.25 cm
Tube Length	=	53 cm
Fin Pitch	=	4.4 per cm
Fin Thickness	=	0.04 cm
Frontal area	=	53 x 22.5 cm
Fin Length	=	22.5 cm

3.2.3 SELECTION OF EXPANSION DEVICE

The expansion device is one of the basic components of a mechanical refrigeration system. The high pressure liquid from the condenser is fed to evaporator through a throttling device which should be designed to pass maximum possible liquid refrigeration to obtain a good refrigeration effect. The liquid line should be properly sized to have minimum pressure drop.

The throttling device is a pressure reducing device and a regulator for controlling the refrigerant flow. It also reduces the

pressure from the discharge pressure to the evaporator pressure without any change of state of the liquid refrigerant. The most commonly used throttling device is the capillary tube.

In the present case a double capillary tube with the following specifications has been used.

Length of capillary tube	=	174 cm
Tube outside dia	=	0.256 cm

3.2.4 SELECTION OF BLOWER AND FAN

Blower is a device to produce flow of air. These are identified into two general groups centrifugal and axial flow blowers. In the present prototype double shaft one blower and one fan was chosen as that of window airconditioner. The blower feeds the conditioner air through evaporator and fan discharges the hot air or cool air to atmosphere as the case may be.

Following type of blower with the specification was used. It is centrifugal type radial blade blower which draws air through the evaporator/condenser and feeds it to the room.

Specifications:

Speed	1970 RPM
Voltage	230 AC
Motor	250 W

3.2.5 DESIGN OF CONDENSER UNIT

The heat added in the evaporator and compressor to the refrigerant is rejected in condenser at high temperature and pressure. The super-heated refrigerant vapour enters the condenser to dissipate its heat in three stages. First on entry the refrigerant loses its superheat, it then loses its latent heat at which the refrigerant is liquidified at saturation temperature pressure. This liquid loses its sensible heat further and the refrigerant leaves the condenser as a sub-cooled liquid.

There are several methods of dissipating the rejected heat into the atmosphere by condenser. These are water cooled, air cooled or evaporative cooled condensers. In case of present condenser the details are as under:

Number of tubes	=	32
Outside dia	=	1.25 cm
Tube Length	=	56 cm
Tube Material	=	Copper
Fin Thickness	=	0.04 cm
Fin Length	=	23.5 cm

3.2.6 HAND SHUT-OFF VALVES

Since heat pumps work by reversing the flow of refrigerant through the system, for reversal of flow reversing valve, special four-way valve or solenoid or hand shut- off manually operated valves are required. The reversing valves are of many types. Some units

use three-way valves, either manually or electrically operated. These valves have one opening to the compressor, one opening to the condenser and one to evaporator. Two of these valves are needed to operate the unit.

Other units use a four-way valve to reverse the flow. It is operated by the movement of one valve stem. They are popular in small tonnage conditioners such as window units.

For the present system hand shut-off valves were used as these are superior to four-way valves from thermodynamic analysis point of view. The six one way valves were used as solenoid leaked.

Specifications:

Danfoss make

Total number of valves = Six

Four number of ∅ 12.7 mm and Two ∅ 6.3 mm

3.3 FABRICATION OF THE SYSTEM

3.3.1 The present system comprises a compressor,an evaporator, a condenser, a blower and a fan with a motor, capillary tube, wood wool pad, a pump, a water tank, electric control board, pressure gauges, hand shut-off valves, aluminium ducting and shutter door. To make it compact these were assembled as those of a window airconditioner unit (Fig. 3.2).

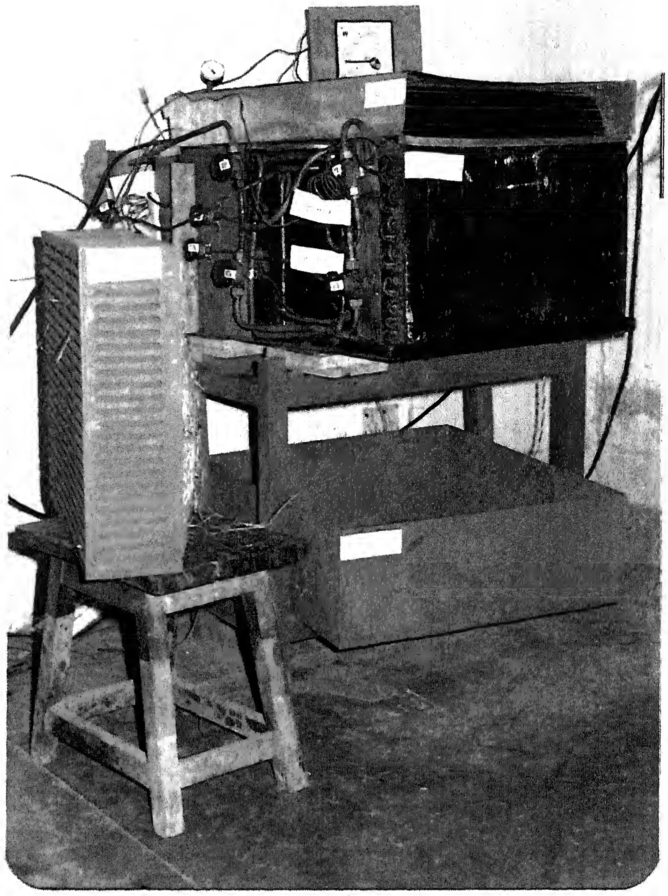

Fig. 3.2. Present developed hybridized all the year-round airconditioning system.

3.3.2 COMPRESSOR

Compressor as per specification given in 3.2.1. has been used for the system. The inlet and outlet connections of the compressor has been brazed to evaporator and condenser coil respectively.

3.3.3 EVAPORATOR

A conventional 1.5 ton evaporator has been used as described in Section 3.2.2.

3.3.4 EXPANSION DEVICE

Two copper cappilary tube having 174 cm each and 0.256 cm outer diameter have been used. The capillary tubes have been connected both to evaporator and condenser coil. It's to be noted that only one set is operational when it's operated in any mode (i.e. heat pump or air-cooling). In the present set, capillary 1 expands the liquid in airconditioning mode and capillary -2 does the expansion when it operates as a heat pump. The capillary No.1 and 2 are connected as shown in Fig. (3.3).

3.3.5 CONDENSER

The condenser unit as per given specification was used.

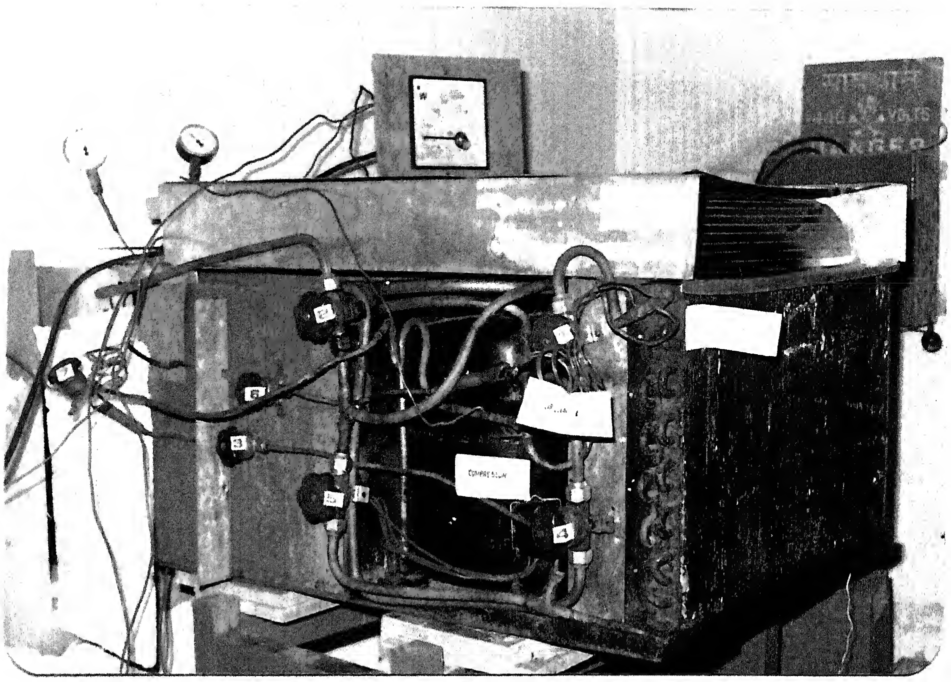

Fig. 3.3. Developed system showing connection of expansion device and other components.

3.3.6 WATER CIRCULATION SYSTEM

(a) A GI tank of 800x700x200 mm size is used for storing and collecting water after being sprayed over cooling pads. The cooling pads are provided on both sides of the openings of the conventional airconditions, Fig. 3.4.

(b) A vertical Tullu pump is used for circulation of water from tank to the spray pipe. The pump is capable of delivering 200 Lpm at a heat of 1.25 m. This head is sufficient for the present system, Fig. 3.

3.3.7 BLOWER, FAN AND DUCTING

Blower and fan has been fitted as that of a normal window airconditioner. The fan sucks the air through the wetted pads and since the discharge to condenser is closed with help of a shutter, so in the evaporative cooling mode, the same air is sent back to room through a designed duct. The blower recirculates the conditioned or cool air and thus higher velocity discharge enhances comfort.

In airconditioning mode blower draws air over the evaporator coil and supplies to the room after cooling and dehumidification. Duct size and dimensions have been calculated and suitable ducting of 1 mm aluminium sheet has been used.

3.3.8 VALVES (ONE WAY)

Several different types of special reversing valves are used in heat pumps. In our case, refrigerant flow is manually reversed and thus six one-way valves were used. The special reversing valves may be operated automatically, manually or electrically (through solenoids).

3.3.9 INSTRUMENTS USED

Instruments used in the system are as under:

(a) Pressure gauge (0 - 300 PSI)

(b) Wattmeter (0 - 5000 W)

(c) Thermometer (- - 120 F, 0 - 500, 0 - 100 C)

(d) Psychrometer (0 - 100 C)

(e) Anemometer (Vane type, 0 - 1,00,000 ft)

Controls used in the system are as below:

(a) Valves for low and high pressure connection to the pressure gauge.

(b) Valves for charging and reversal of refrigerant in the system.

(c) Switches for operating compressor, blowers and pump.

3.3.10 ELECTRICAL CIRCUIT

A 230 V single phase stabilized electric supply was given to the switch board of the system. The power was distributed to compressor, blowers and pump.

3.4 SELECTION OF REFRIGERANT

Refrigerant-22 ($CH\ Cl\ F_2$) and two different mixtures of refrigerant-12 and refrigerant-22 were used to see the effect on performance and performance index.

3.5 MISCELLANEOUS

In order to reduce the heat gain/loss to the cool air as the case may be 2.54 cm thick thermocole was also wrapped round the supply and distribution ducts. The M.S. sheet grilles are provided in the suction line. The M.S. sheet discharge grilles are adjusted to give uniform air distribution as well as spread of air as per desire. The velocity of air at grilles was kept at 4 - 5.5 m/s.

3.6 PRESSURE TEST

After installation is completed and all electrical checks have been made, the system was pressurized to 250-300 PSIG using dry

Nitrogen. Leak was tested using soap solution and in case of any leak it was removed by brazing and tightening of joint. When all joints are tested satisfactorily,the pressure was released allowing the gas to escape.

3.7 EVACUATION AND CHARGING

After it was known that the system was leakproof, all the air and moisture were removed. Air was pumped out of the system through the charging line with a vacuum pump. Thus the system was evacuated to almost zero vacuum and it was tested for leakage under vacuum for 24 hr duration. Thatafter, the system was purged with R-22 and evacuated twice such that the system is free from air and other non-condensable gases. Finally, the system was charged with R-22 for the first set and then with mixture of R-12 and R-22, such that rated current is taken by refrigeration system. The head pressure was also checked during this process in addition to current taken by the system. Subsequently, the system became ready for the operation.

3.8 SYSTEM OPERATION

3.8.1 EVAPORATIVE COOLING MODE

For the months of April to June and October we need the process of cooling by humidification. The present system can be

operated in that mode. It is worth mentioning that the use of the conventional airconditioner is rather not correct.

The system can be brought to operation as under:

a. Water is sprayed over the pad with the help of a pump.

b. Subsequently, the fan at the condenser side is switched on for sucking of air through the wetted pads where it gets evaporatively cooled.

c. The cool air due to humidification is supplied to the room, when the condenser side is closed by the shutter door.

d. The blower at evaporator side recirculates the air from the room in order to discharge with the higher grille velocity.

e. Higher air circulation can be used in the room if the ambient condition is too severe.

The above-mentioned mode of operation provides thermal relief for ›t-dry climates.

3.8.2 AIRCONDITIONING MODE

The air needs to be cooled and dehumidified for the months of July to September. This is achieved by the conventional air-conditioner.

The present system operates as an airconditioner as described below: When valves numbered 1,2,3 as shown in Fig are open.

(a) The blower and fan are switched on and then compressor is also put into operation.

(b) The low-temperature, pressure, superheated vapour from evaporator conveyed through the suction line is compressed by compressor to a high pressure.

(c) Heat added in the evaporator and compressor is rejected in condenser at high temperature and pressure.

(d) Liquid refrigerant from condenser at high pressure is fed through a throttling device to an evaporator at low pressure.

On absorbing the heat to be extracted from media to be cooled, the liquid refrigerant boils actively in the evaporator and changes state.

For the present system the valves No. (1,2,3) as shown in Fig. (3.1) are operational. The centrifugal radial vane blower sucks the air from the room and discharges through the cooling cools of evaporator on grille. The fan blows off the atmospheric air over the condenser and thus the condenser gets cooled discharging heat to the atmosphere.

3.8.3 HEAT PUMP MODE

Space heating can be supplied by different systems. Heat pump system is one of them and the same is superior to the duct heaters or electric resistance heat.

Heat pump is remarkable for it can move heat in two directions. Heat pumps work by reversing the flow of the refrigerant through the system. In a heat pump system, each heat transfer coil is used both as an evaporator or a condenser coil, depending on whether the system is in the heating or cooling mode.

In the present, system, hand valves were used to reverse the cycle which was described in 3.8.2. Here travel of refrigerant is reversed to change from cooling to heating.

The operational procedure is the same as that of a/c mode of operation. Here the valve Nos. [4,5,6] as shown in Fig. (3.1) are operational and evaporator is converted to condenser and vice versa. Same airconditioner unit operates as heating unit, releasing heat to the room through condenser coil and grille.

CHAPTER-4

RESULTS AND DISCUSSIONS

Various quantities viz. suction and delivery air conditions, evaporator and condenser pressures, compressor power have been obtained and tabulated. Comparison have been made between heat pump and air-conditioning modes of operation on the basis of COP and PI.

A general lay-out for larger capacity plant has been envisaged and economic analyses have been done in order to emphasize the usefulness of the present development.

4.1 EVAPORATIVE COOLING MODE OF OPERATION

The present hybridized system has provision for operation in the evaporative cooling mode. For this case the volume of evaporatively cooled air to the room was measured to be 1440 m^3/hr.

Volume of the room = (4.24 x 4.24 x 4.30) m^3 = 77.3m^3

Hence the number of air changes per hour is found to be 13.6, which is an accepted and most satisfactory value.

4.2 THERMODYNAMIC ANALYSIS FOR VAPOUR COMPRESSION CYCLE

For the simple vapour-compression cycle variations of the heat rejection, refrigeration effect, compressor power and COP were found out for R-22 for different operating conditions. Besides, calculations were performed considering the pressure drops during suction and discharge of compressor, superheat of vapour in the evaporator, subcooling of condensate to get the results close to that of an actual operating vapour compression system.

The various characteristics are as follows:

a. Heat rejection to condenser

It reduces slowly with rising condensing temperature because of the reduction in mass flow rate of refrigerant due to decrease in volumetric efficiency. The results are shown in Fig. 4.1 for different evaporator temperatures.

b. Refrigeration capacity ($\dot{Q}_c$)

Its variation with condensing temperature for different evaporator temperatures is shown in Fig. 4.2. It also decreases at faster rate due to decrease in mass flow rate and refrigeration effect.

c. Compressor power

It has been exhibited in Fig. 4.3 with varying condensing temperatures and for different evaporator temperatures. Power increases with increasing condenser temperature since compressor has to compress the refrigerant to the higher

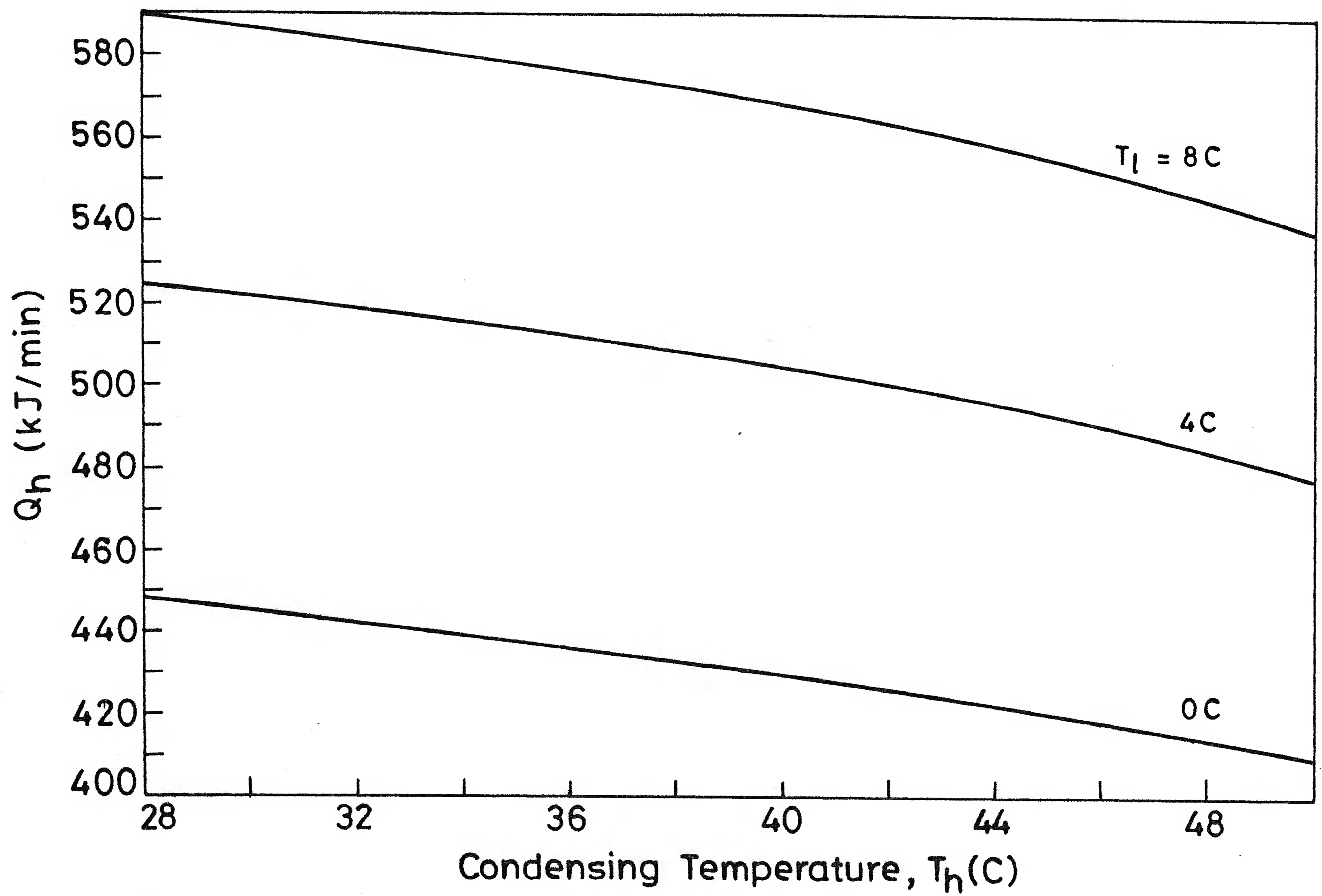

Fig. 4.1 Variation of heat rejection with condensing temperature.

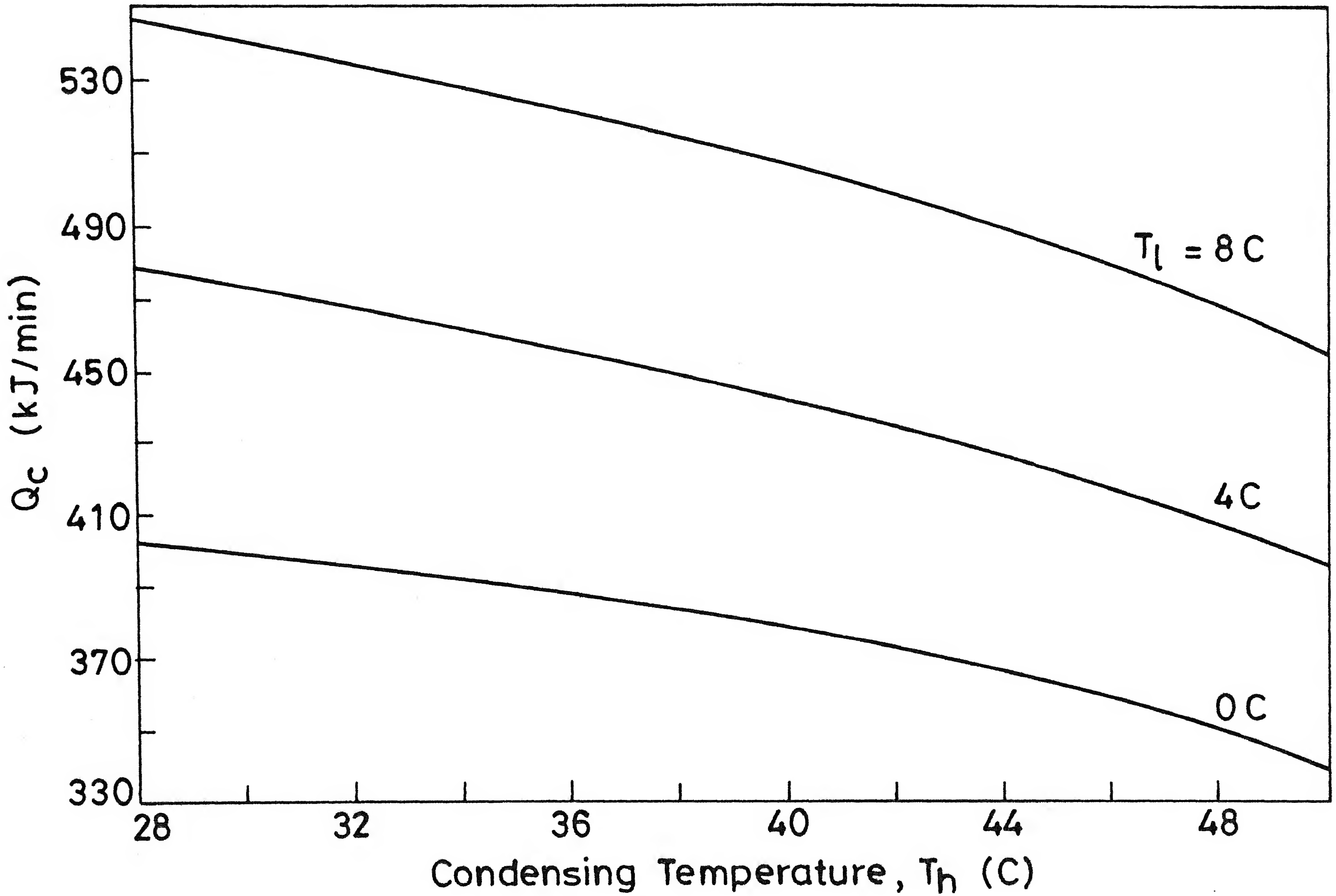

Fig. 4.2 Variation of refrigeration effect with condensing temperature.

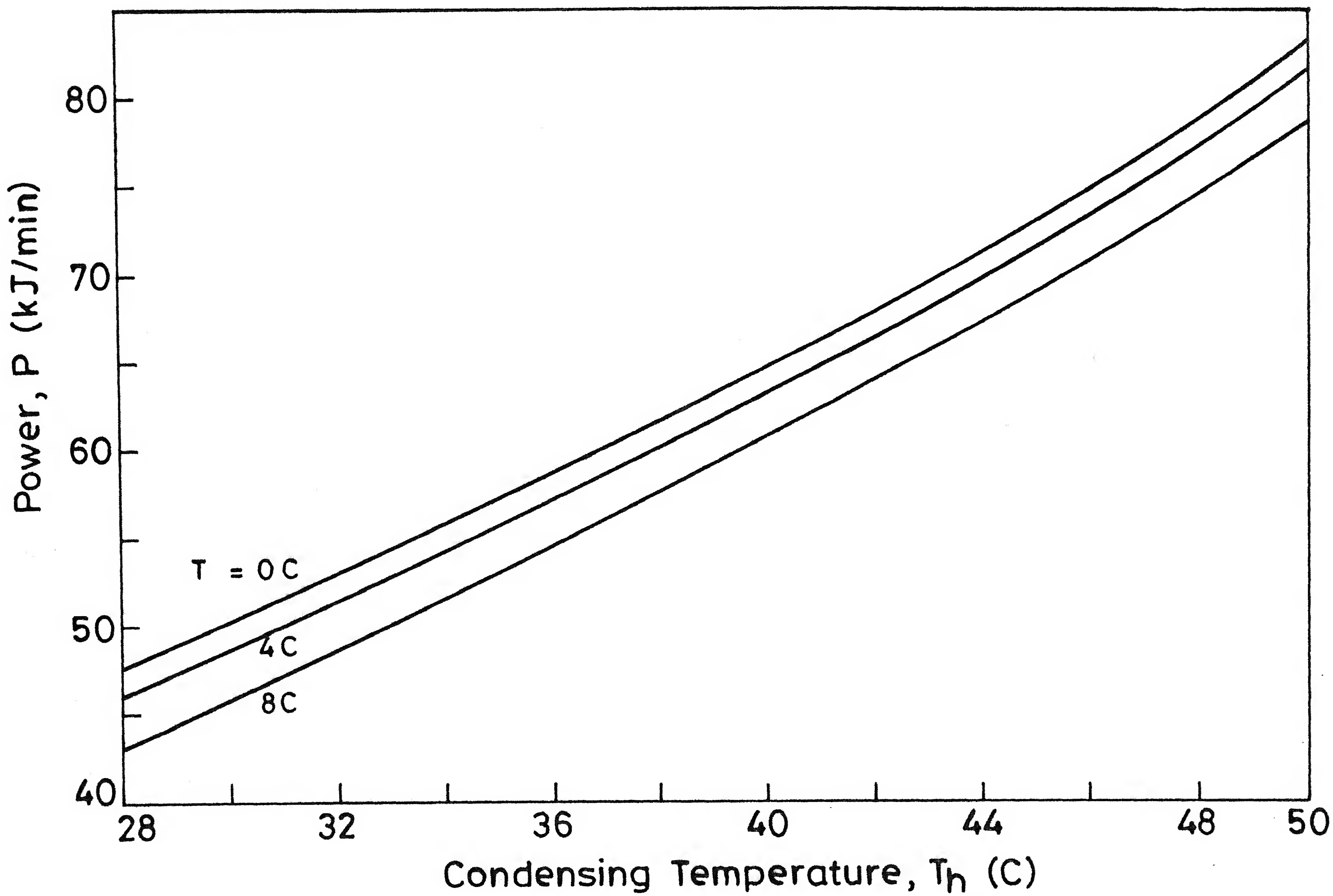

Fig. 4.3 Variation of power with condensing temperature.

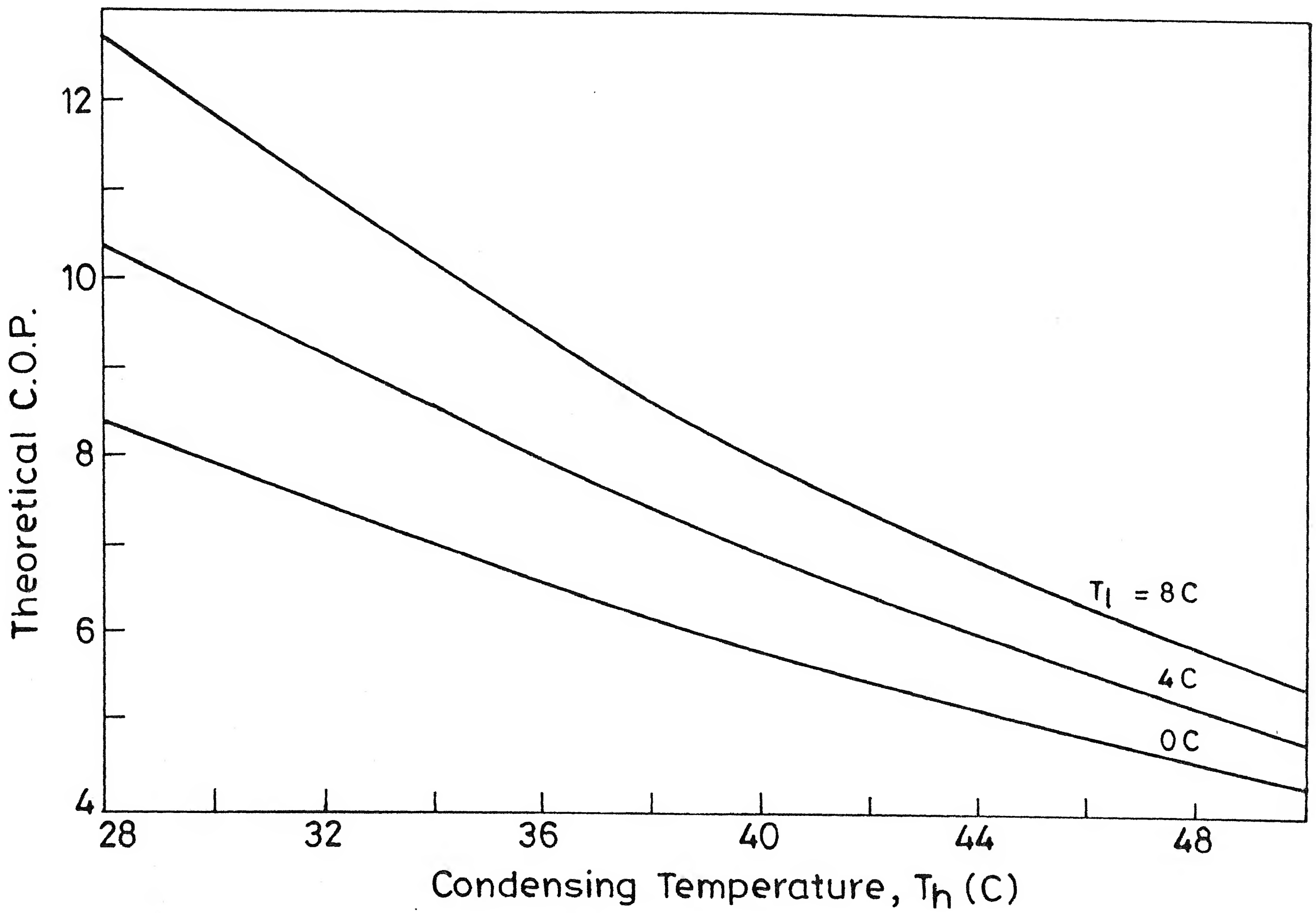

Fig. 4.4 Variation of theoretical C.O.P. with condensing temperature.

pressure. Hence even though mass flow rate decreases still the overall power requirement is more.

d. COP

COP is found to decrease as $\dot{Q}_c$ decreases and power input increases with corresponding increase in condensing temperatures. This is also found for different evaporator temperatures.

Fig. 4.4 depicts the above variation.

4.3 EXPERIMENTAL RESULT OF HEAT PUMP/ REVERSE CYCLE OPERATION

Tables 4.1-4.4 show the experimental data obtained during the months of January and February, 1990 at the time of operation of the system. For above experimental result, our system was charged separately with the following refrigerants.

(i) R-12 (1.2 kg)

(ii) R-22 (0.350 kg)

(iii) mixture of R-12 and R-22

a) R-12 : R-22 :: 41% : 59% by weight

b) R-12 : R-22 :: 51% : 49% by weight

From the tables 4.1-4.4 last three steady values were taken for the calculations of COP and PI were calculated for heating produced in the room and cooling produced in the surroundings.

Table 4.1

Experimental Results for Reverse Cycle Operation Using R-12(1.2 kg)

Sl. No.	CONDENSER				EVAPORATOR				POWER CONSUMPTION		PRESSURES		
	Suction air		Air Entering room		Ambient conditions		Exhaust		Compressor (W)	Blower motor (W)	Evaporator (bar)	Condenser (bar)	Pressure Ratio
	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}					
1	23.5	20.3	27.5	21.7	20.5	19	20	18.5	1170	250	2.737	12.0474	4.4016
2	25.4	21	28.5	22.5	20.3	18.8	19	17.5	1180	250	2.737	12.0474	4.4016
3	25.2	21	28.2	22.2	20.0	18.7	19.5	17.7	1180	250	2.944	12.0474	4.092
4	25.2	21.1	28.8	22.0	19.8	18.5	18.8	17.2	1180	250	2.8865	12.5992	4.3648
5	26.2	21.4	31.5	23.5	19.5	18.0	18.3	16.8	1190	250	2.93	12.5992	4.3
6	26.5	21.5	33.5	24.0	19.3	17.8	17.5	16.2	1200	250	3.08	12.7371	4.1328
8	26.8	21.5	36	25.2	19.0	17.5	17.0	15.5	1220	250	3.0819	12.7371	4.1328
9	27.0	21.5	36	25.2	19.0	17.0	16.5	15.2	1220	250	3.0819	13.0819	4.244
10	27.0	21.5	36	25.2	19.0	17.0	16.5	15.2	1220	250	3.0819	13.0819	4.244

Table 4.2

Experimental Results for Heating Mode Operation Using R-22 Alone (0.85 kg)

SL. NO.	CONDENSER				EVAPORATOR				POWER CONSUMPTION		PRESSURES		
	Suction air		Air Entering room		Ambient conditions		Exhaust		Compressor (W)	Blower Motor (W)	Condenser (bar)	Evaporator (bar)	Pressure ratio
	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}					
1	27.4	21.2	34.5	23.6	21	19	18.5	15.7	1340	250	5.1509	16.873	3.275
2	27.3	21.4	35.0	23.8	20.9	18.4	17.8	14.7	1350	250	5.1509	17.57	3.411
3	27.3	21.2	35.8	24.4	20.5	18.0	17.0	14.9	1360	250	5.2888	18.25	3.4512
4	27.4	21.4	36.2	25	20.2	17.7	16.5	14.5	1360	250	5.4957	18.2543	3.3215
5	27,3	21.0	36.5	25	19.5	17.5	16.2	14.2	1360	250	5.4957	18.944	3.447
6	27.5	21.6	37.5	25	19.0	17.0	16.0	14	1340	250	5.6336	19.6335	3.48
7	27.5	21.6	38.7	28.4	19.0	17.5	15.2	14.2	1350	250	5.6336	19.6335	3.48
8	27.5	21.6	38.7	28.4	19.0	17.5	15.2	14.2	1350	250	5.63	19.6335	3.48

Table 4.3

Experimental Results for Heating Mode Operation Using Mixture of R-12 and R-22 (Total wt 1.1 kg) in the ratio of 41% : 59%

SL. No.	CONDENSER				EVAPORATOR				POWER CONSUMPTION		PRESSURES		
	Suction air		Air Entering room		Ambient conditions		Exhaust		Compressor (W)	Blower Motor (W)	Condenser	Evaporator	Pressure ratio
	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}					
1	24.8	19.0	31	20	21	19	18.5	15.5	1160	250	15.4957	5.15	3.008
2	25	19.0	32	20.6	20.8	18.5	17.5	14.9	1180	250	16.1854	5.15	3.142
3	25.2	19.0	33	21.5	20.5	18.0	16.5	15.0	1200	250	16.8751	5.495	3.07
4	25.5	19.2	34.5	22	20.0	17.5	16.5	15.0	1220	250	17.563	5.495	3.196
5	26.0	19.5	36.5	22.8	19.5	17.5	16.5	14.8	1240	250	17.9095	5.6336	3.17
6	26.7	19.7	38	24	19.0	17.2	16.3	14.9	1280	250	18.253	5.6336	3.24
7	27.0	20.0	39	24.6	19.0	17.2	16	14.5	1270	250	18.2543	5.84	3.12
8	27.5	21	39.5	24.2	19.0	17.0	16	14.8	1290	250	18.5992	5.8405	3.1845
9	27.5	21.2	40.5	25.5	19.0	17.0	16	14.8	1300	250	18.944	5.8405	3.2435
10	27.5	21.2	40.5	25.5	19.0	17.0	16	14.8	1300	250	18.944	5.8405	3.2435

Table 4.4

Experimental Results using R-12 and R-22 in the Ratio of 51:49

SL. No.	CONDENSER Suction air T_{db}(c)	T_{wb}(c)	Entering air T_{db}	T_{wb}	EVAPORATOR Ambient conditions T_{db}	T_{wb}	Exhaust T_{db}	T_{wb}	POWER CONSUMPTION Compressor (W)	Blower Motor (W)	PRESSURES Condenser	Evaporator	Pressure Ratio
1	25.2	19.0	31	23	20.8	19.2	19.5	16.8	1450	250	16.8751	2.737	6.165
2	25.4	19.0	33.5	23.5	20.5	19.0	19.5	16.5	1470	250	17.57	2.944	5.96
3	25.8	19.2	36.2	24.2	20.5	19.0	19.0	16.5	1520	250	18.944	3.2198	5.88
4	26.2	19.5	37.4	24.5	20.3	18.8	19.0	16.5	1540	250	19.6336	3.495	5.617
5	26.2	19.6	37.8	25.0	20.3	18.8	18.8	16.4	1560	250	19.7716	3.495	5.65
7	26.3	19.8	38	25.2	20.0	18.5	18.5	15.8	1570	250	19.7716	3.634	5.657
8	26.4	20.0	38.5	25.5	19.8	18.5	18.5	15.5	1570	250	19.6336	3.495	5.617
9	26.5	20.8	39	25.8	19.6	18.3	18.2	15.4	1580	250	19.9785	3.7716	5.29
10	26.5	21.5	39.5	26.0	19.5	18.2	18.0	15.8	1600	250	19.6336	4.461	4.4011
11	26.5	21.5	39.8	26	19.5	18.2	18.0	15.5	1600	250	19.9785	4.8061	4.156
12	26.5	21.5	39.8	26	19.5	18.2	18.0	15.5	1600	250	19.9785	4.8061	4.152

Table 4.5

Sl. No.	$\dot{Q}_h$ kW	$\dot{Q}_c$ kW	$P = \dot{Q}_h - \dot{Q}_c$	P_{actual} kW	Refrigerant used (Amount kg)	COP	PI
1	4.624	3.088	1.536	1.48	R-12 (1.2)	2.12	3.12
2	5.664	3.978	1.686	1.6	R-22 (0.85)	2.486	3.54
3	4.98	3.099	1.481	1.54	R-12 and R-22 in the ratio 41:59 (1.1)	2.044	2.97
4	5.23	3.277	1.953	1.85	R-12 and R-22 in the ratio 51 : 49 (1.1)	1.68	2.79

Table 4.6

Experimental Results on Airconditioning Mode using Mixture of R-12 and R-22 [case IV]

Sl. No.	EVAPORATOR				CONDENSER				POWER CONSUMPTION		PRESSURES		
	Suction		Discharge to room		Ambient conditions		Exhaust		Compressor	Blower	Condenser (bar)	Evaporator (bar)	Pressure Ratio
	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}	T_{db}	T_{wb}					
1	21.2	20	18	16.5	24	20.7	24	20.5	1400	250	15.495	4.46	3.47
2	20.5	19	17.5	16	23.7	20.5	25	21	1400	250	15.495	4.46	3.47
3	18.0	17.5	17	15.5	23.5	20.6	26	21.8	1400	250	16.185	4.46	3.6289
4	17	16	15.5	14.2	23.2	20.6	26.5	22	1400	250	16.185	4.46	3.628
5	16.8	15.8	15	14	23	20.2	27	22.4	1400	250	16.875	4.59	3.676
7	16.5	14.5	14.8	13.5	23	20.2	27.7	22.4	1400	250	17.013	4.59	3.7065
8	16	14.5	15	14	23	20.2	28	22.5	1400	250	17.013	4.59	3.7065
9	16	14.5	15	14	23	20.2	29	22.5	1400	250	17.564	4.806	3.655
10	16	14.5	15	14	23	20.2	30.5	23.5	1400	250	16.875	4.806	3.511
11	16	14.5	15	14	23	20.2	30.5	23.5	1400	250	16.875	4.806	3.511

$\dot{Q}_h$ = 4.158 kW $\dot{Q}_c$ = 2.44 kW P = 1.65 kW COP = 1.48 PI = 2.52

4.4 COMPARISON BETWEEN HEAT PUMP AND AIR-CONDITIONING CYCLE

The system operated in air-conditioning mode take $P = 1.65$ kW, $\dot{Q}_c = 2.442$ kW and $\dot{Q}_h = 4.153$ kW thereby. In this case COP and PI are found on the basis of cooling produced inside the room and heat rejection from condenser to surrounding.

For the comparison we chose the mixture of refrigerant R_{12} and R_{22} in the ratio of 51% and 49% by weight. In the heat pump mode the corresponding values of P, $\dot{Q}_c$ and $\dot{Q}_h$ were found to be 1.35, 3.277 and 5.23 kW respectively. Seemingly, the same mixture gives more refrigeration effect $\dot{Q}_c$ and heat rejection when it operates as heat-pump. Evidently the system shows that the system is not performing in both modes in the ratio of 1:1. This is due to the fact that the condenser and evaporator are of different capacities.

4.5 FIVE, TEN AND HUNDRED TON HYBRIDIZED ALL THE YEAR-ROUND AIR CONDITIONING SYSTEM

Realizing the usefulness of a small capacity hybridized all the year-round air conditioning system, energy and economic analyses are done for 5 ton, 10 ton and 100 ton cooling capacity plant. Such units provide larger leeway to manoeuvre them in a suitable duct net-work. Fig. 4.5 shows the schematic lay-out of the same. The dampers are provided at various positions in the duct in order to monitor the supply and discharge of air as per requirement. The provisions are made to operate the system in following ways:

a. Evaporative cooling mode for hot-dry climates

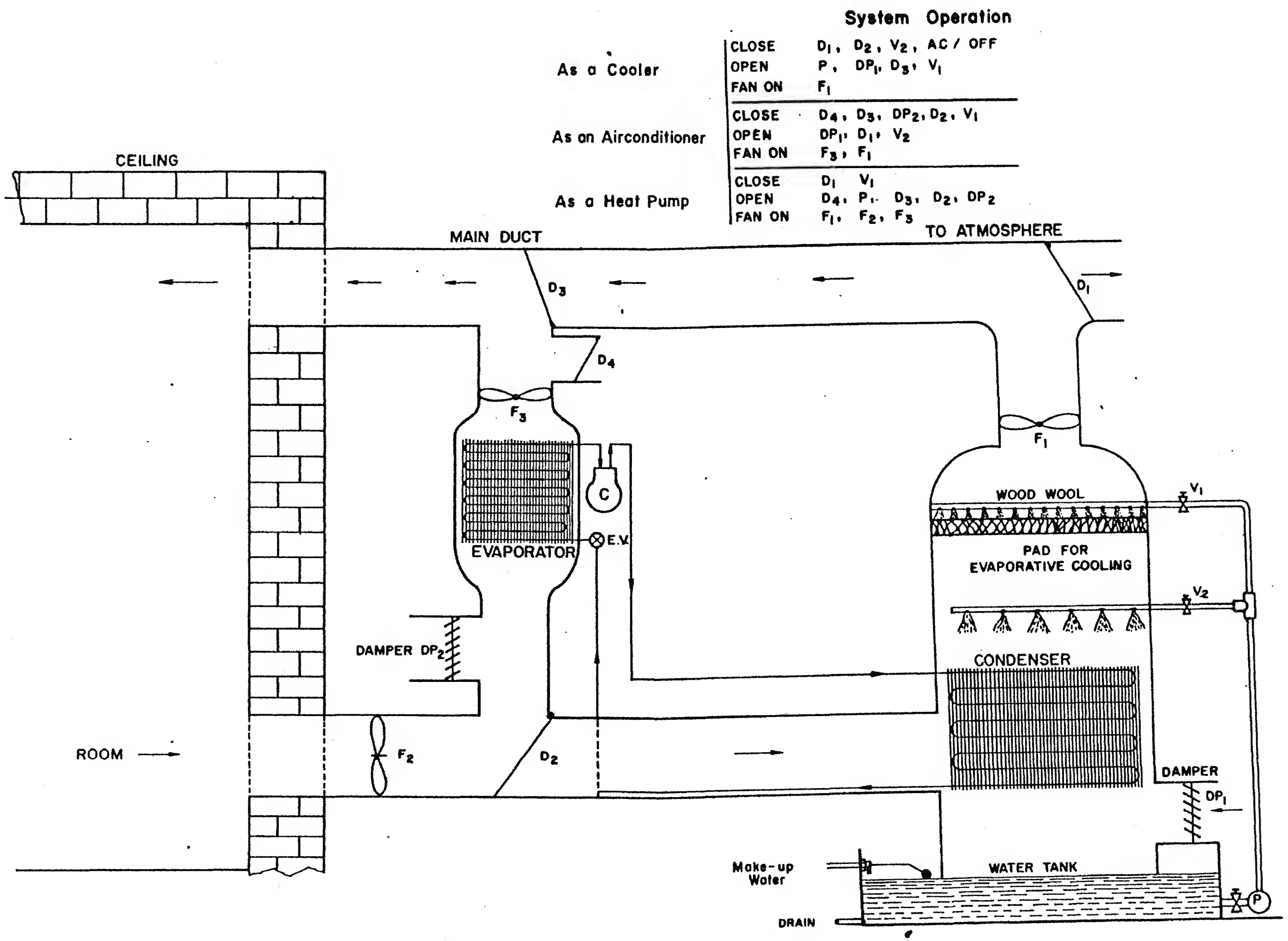

Fig.4.5 Schematic layout of a hybridized all the year round air conditioning system for larger capacity plant.

b. Air conditioning mode for the hot-humid climates

c. Reverse cycle operation for the winter-heating.

The manipulation of damper and values rests with the user. If the user does not get satisfied by a particular mode of operation, he may operate the system in appropriate mode to obtain comforts. However he should make a compromise between energy and comfort as far as practicable. Figure 4.5 also provides valves and dampers positions for various modes of operation. The analyses are done in respect of energy saving and reduction in running electricity cost.

Table D. gives the typical cooling load for 5 ton, 10 ton and 100 ton capacity units. For all these capacity plants calculations have been carried out for taking COPs as 2 and 3 of the system and conclusions have been drawn based on energy saving and cost of energy. Detailed calculations are given in Appendix D. The energy-saving per year and annual electricity costs for 5 ton, 10 ton and 100 ton units are given in Table 4.7.

Table 4.7

Energy and Cost for 5, 10, 100 Ton system

	5 TON PLANT			10 TON PLANT			100 TON PLANT		
	Conventional System	*Modified System	Difference %	Conventional System	*Modified System	Difference	Conven-tional	*Modified	Diff-erence
Energy Per year (kWh/year)	20,260	7563.84	12696.16 (62.66%)	40520	15127.68	25392.32	4,05,200	151276.8	253923.2
Cost of electrical bill (Rs/year)	16,208	6051	10,157 (62.66%)	32416	12102	20314	324160	121020	203140

* It stands for the suggested hybridized all the year-round airconditioner.

CHAPTER-5

CONCLUSIONS AND SUGGESTIONS

5.1 CONCLUSIONS

1. Evidently, R-22 is seen to be the best. The advantage in the use of R-12 or the mixture lies in that the compressor runs cooler.

2. The mixture of R-12 and R-22 was found to give better performance in heating mode than conventional airconditioning mode.

3. COP and PI were 1.68 and 2.79 in reverse cycle/heat pump mode of operation and in airconditioning mode respective values came out to be 1.48 and 2.52.

4. For the heat pump operation the present 1.5 ton unit took 1480, 1600, 1540 and 1850 W when refrigerants were R-12, R-22, a mixture of R-12 and R-22 in the ratio of 41% : 59% and 51% : 59% by weight respectively.

5. Evaporatively cooled air that could be delivered to the room was measured out ot be 1440 m^3/hr which is sufficient for the required room.

6. The larger unit has wide scope of arrangement of various components in a better way as compared to smaller unit. Hence larger unit will be more compact smaller unit, as seen from our experience in the fabrication of the same.

7. Saving in energy and running cost per annual basis for larger unit plant (100 ton unit) is estimated to be little over 2 lakhs. This is a very significant saving (62.6%) and hence vouchsafes our new venture.

5.2 SUGGESTIONS

1. More number of such systems need to be fabricated and tested for the year-round operation and should be operated atleast for a few years in order to explore the nationwide venture of developing such system.

2. The higher tonnage unit has better prospects due to greater flexibility of space for arrangement of various unit. The smaller unit becomes rather bulkey compared to conventional window airconditioner unit and may not appeal to some of the users.

3. To make the operation easier, soleniod valves should be used, and all the control points should be so fitted that they can be operated from the front panel.

4. The present type of 3-in one should be tried in some Government and private offices and opinion poll/survey be collected for the nation-wide publicity of the new development.

The afore-said suggestions would go a long way making our present hybridized all theyear-round airconditioning system a most desirable unit for the buyers.

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APPENDIX-A

THERMODYNAMIC ANALYSIS OF VAPOUR-COMPRESSION CYCLE BASED ON THE ACTUAL CYCLE

Assuming T_{sup} = 20 C, subcooling 6°C. Compression, depression and wire drawing equivalent to 4 C and taking

$$T_h = 45 \text{ C}, \; T_1 = 6\text{°C}$$

From superheated tables,

$$h_1' = 268.0 \; \frac{kJ}{kg} = h_1''$$

(assuming $h_1' - h_1''$ to be isentholpic process)

$$S_1'' = 0.9798 + \frac{1.0039 - 0.9798}{273.58 - 266.348} \times (268 - 266.348)$$

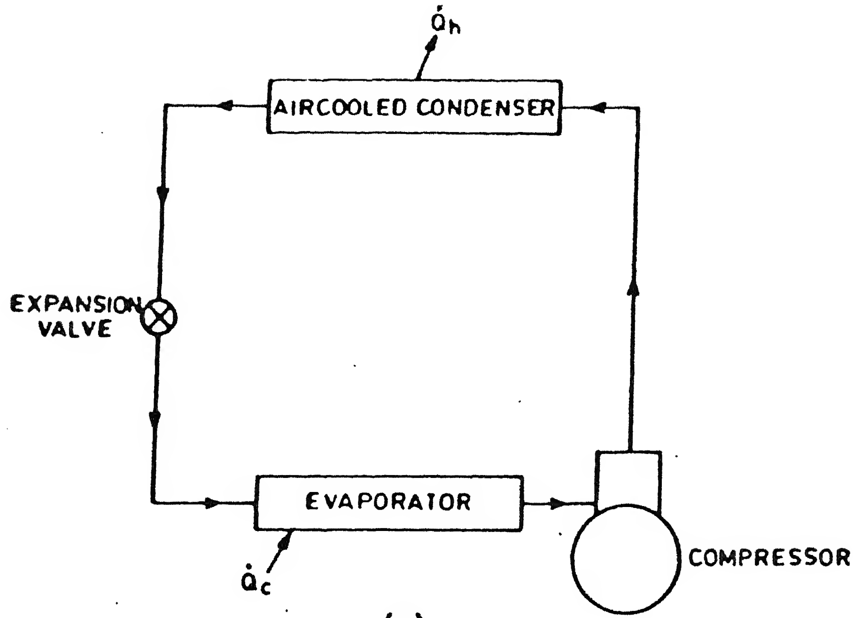
$$= 0.9853 \text{ kJ/kg-K} = S_2'$$

Assuming process 1" - 2' to be isentropic

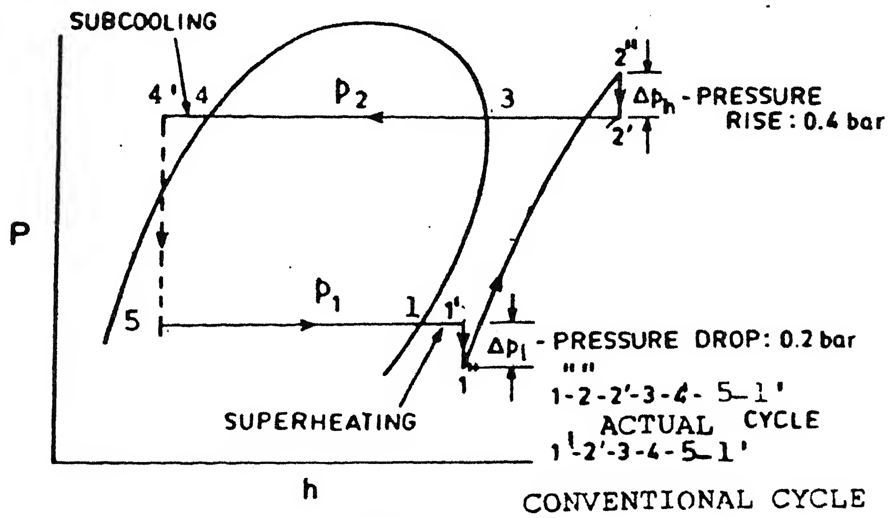
$$h_2' = 298.66 + \frac{307.31 - 298.66}{1.0022 - 0.9781} \times (0.9853 - 0.9781)$$

$$= 301.246 \; \frac{kJ}{kg}$$

$$h_4' = 94.14 \text{ kJ/kg}$$



(a)



(b)

Fig. Single-stage vapour-compression system
P-h diagram for single-stage vapour-compression cycle.

$$\text{Pressure ratio} = \left(\frac{18.889}{5.311}\right) = 3.5565$$

Compressor efficiency for R-22 system is given by

$$c = \frac{1}{1.13023 + 1.13289 \times 10^{-1} Pr - 3.334529 \times 10^{-2} Pr^2 + 4.86757 \times 10^{-3} Pr^3 - 2.134 \times 10^{-4} Pr^4}$$

$$= 0.76873$$

$$h_2'' = \frac{h_2' - h_1''}{c} + h_1''$$

$$= 311.2479 \text{ kJ/kg}$$

Considering 1.5 ton system

$$\dot{m} = \frac{1.5 \times 3.5}{h_1' - h_5} = \frac{1.5 \times 3.5}{268 - 94.14} = 0.030196 \text{ kg/s}$$

$$= 1.8118 \text{ kg/min}$$

$$\dot{Q}_h = \dot{m}\,(h_2'' - h_4')$$

$$= 0.030196\,(311.2479 - 94.14)$$

$$= 6.5557 \text{ kW}$$

APPENDIX -B

THERMODYNAMIC ANALYSIS OF VAPOUR COMPRESSION CYCLE BASED ON EXPERIMENTAL DATA

Evaporator pressure = 5.4957

Condenser pressure =18.944

By interpolation, we obtain from superheated tables, corresponding evaporator temperature

$$= \quad 2 + \frac{2}{5.65 - 5.31} \times (5.4957 - 5.31)$$

$$= \quad 3.09 \text{ C}$$

Similarly condenser temperature

$$= \quad 48 + \frac{50 - 48}{19.327 - 18.458} \times (18.944 - 18.458)$$

$$= \quad 49.12 \text{ C}$$

Assuming evaporator depression to be 1.09 C and compressor depression to be 2.09 C, with 20 C superheat

$$h_1' = 266.348 + \frac{267.18 - 266.348}{2} \times 1.09 = 266.801 \frac{kJ}{kg}$$

$$= h_1'' \text{ (Assuming } h_1' - h_1'' \text{ to be isenthalpic)}$$

$$s_1'' = 0.9798 + \frac{1.0039 - 0.9798}{2.7358 - 266.348} \times (266.801 - 266.348)$$

$$= 0.98131 \text{ kJ/kg-K}$$

Assuming process 1" -2' to be isentropic and interpolating the values of h and s corresponding to a temperature, of 49.12 C.

$s_{49.12}$ at 40 C superheat

$$= 0.9755 + \frac{0.9731 - 0.9755}{2} \times (1.12)$$

$$= 0.97415$$

$s_{49.12}$ at 50 C superheat

$$= 0.9995 + \frac{0.9971 - 0.9995}{2} \times 1.12$$

$h_{49.12}$ at 40 C superheat

$$= 299.61 + \frac{300.25 - 299.61}{2} \times 1.12$$

$$= 299.966 \text{ kJ/kg}$$

$h_{49.12}$ at 50 C superheat

$$= 308.33 + \frac{309.06 - 308.33}{2} \times 1.12$$

$$= 308.73 \text{ kJ/kg}$$

Hence, by interpolation

$$h_2' = 299.966 + \frac{308.73 - 299.966}{0.99815 - 0.97415} \times (0.98131 - 0.97415)$$

$$= 302.58 \text{ kJ/kg}$$

Pressure ratio $Pr = \dfrac{18.944}{5.4957} = 3.447$

Compressor efficiency $= 0.7724$

$$\therefore \quad h_2'' = h_1'' + \frac{h_2' - h_1''}{c}$$

$$= 266.801 + \frac{302.58 - 266.801}{0.7724}$$

$$= 313.12 \text{ kJ/kg}$$

Assuming 6C subcooling of condensate

h_4' for 43.12 C

$$= 99.21 + (100.48 - 99.21) \times 0.12$$

$$= 99.362 \; h_5$$

$$\dot{m} = \frac{1.5 \times 3.5}{h_1' - h_5}$$

$$= \frac{1.5 \times 3.5}{266.801 - 99.362} = 0.0313 \text{ kg/s}$$

$$= 1.88 \text{ kg/min}$$

Heat rejection

$$= \dot{m}\,(h_2'' - h_4')$$

$$= 0.0313\,(313.12 - 99.362)$$

$$= 6.69 \text{ kJ/sec} = 6.69 \text{ kW}$$

Power of compressor

$$= \dot{m}\,(h_2'' - h_1'')$$

$$= (313.12 - 266.801) \times 0.0313$$

$$= 1.4497 \text{ kW}$$

APPENDIX- C

SAMPLE CALCULATIONS OF EXPERIMENTAL RESULTS

The air flow rates across evaporator and condenser were taken using a calibrated vane-type anemometer. The state points of air entering and leaving evaporator and condenser were marked on the psychrometric chart. State pts. as reached finally were taken from table Nos. 4.1, 4.2, 4.3 and 4.4.

R-12 table 4.1

We know $\dot{m}_{ac} = \frac{A \times V}{\nu}$

Where

$\dot{m}_{ac}$ - Mass flow through evaporator kg/sec.

A - Frontal area - m^2

V - Velocity of discharge air x m/min

ν - Specific volume kg/m^3

For the present system

$A = 0.6 \times 0.34\ m^2$

$V = 1.08$ m/min

$\nu = 0.8325$ (from Psychrometry chart)

$$\dot{m}_{ac} = \frac{0.6 \times 0.34}{0.8325} \times \left(\frac{108}{60}\right) = 0.44108 \text{ kg/sec.}$$

h_1 ($T_{db} = 19.0$) = 47 kJ/kg
($T_{wb} = 17.0$)

h_2 ($T_{db} = 16.5$) = 40 kJ/kg
($T_{wb} = 15.2$)

Similarly

$$\dot{m}_{ac} = \frac{A_d x V_d}{\nu}$$

Where,

$\dot{m}_{ac}$ = Mass flow rate through duct in kg/sec.

A_d = Area of duct m^2

V = Velocity through duct m/sec.

ν = Specific volume (kg/m^3)

For our system, $A_{duct} = 0.525 \times 0.11 \; m^2$

$V_d = 290 \text{ m/min} \times \frac{1 \text{ min}}{60 \text{ sec}}$

ν = 0.845 from psychrometry chart

$$\dot{m}_{ac} = \frac{A_d x \; V_d}{\nu}$$

$$= \frac{0.525 \times 0.11}{0.845} \times \frac{290}{60} = 0.3302 \text{ kg/sec.}$$

h_3 (T_{db} = 30 C T_{wb} = 25.2 C) = 76 kJ/kg,

h_4 (T_{db} = 27 C T_{wb} = 21.5 C) = 62 kJ/kg

$$\dot{Q}_h = \dot{m}_{ac} \times (h_3 - h_4) = 0.3302 \times (76 - 62) \frac{kJ}{kg} \times \frac{kg}{sec}.$$

We know from energy balance, equation $\dot{Q}_h - \dot{Q}_e = P$

Hence P = 4.624 kW - 3.087 kW = 1.537 kW

Where $Q_e = 0.44108\,(47-40) = 3.087$ KW.

For our system, from actual readings

$$P = P_{compressor} + P_{fans} = 1220 \text{ W} + 250 \text{ W}$$
$$= 1470 \text{ W}$$
$$= 1.47 \text{ kW}$$

$$COP = \frac{\dot{Q}_e}{P} = \frac{3.087}{1.47} = 2.1$$

(coefficient of performance)

$$PI = \frac{4.624}{\cancel{3.087}\ 1.47} = 3.14$$

(performance index)

Based on Power from energy balance, PI is seen to be $\frac{4.624}{1.537} = 3.008$.

APPENDIX - D

CALCULATION

A COP is assumed to be 2 as per normal value.

$$COP = \frac{\text{Refrigeration effect}}{\text{Power}}$$

Summer cooling load for 5 ton capacity is 17.5 kW (Table D)

$$\therefore COP = \frac{17.5}{P}$$

$$P = \frac{17.5}{2} = 8.75 \text{ kW}$$

Conventional airconditioning unit will require 8.75 kW

We represent it by notation $P_1 = 8.75$ kW

Hybridized system will require $\frac{300 \text{ W}}{1.5} \times 5 = 1000 \text{ W} = 1 \text{ kW} = P_1'$ (Say)

For Rainy Season, conventional system will take $\frac{17.5}{2} = 8.75$ kW (COP = 2)

$= P_2$ (Say) for a 5 ton plant

Present system needs $\frac{17.5}{2} \left(\frac{1.65}{2.2}\right) = 6.56 \text{ kW} = P_2'$ (Say)

Factor $\left(\frac{1.65}{2.2}\right)$ has been taken for the special care in the designing stage and subsequent higher energy efficiency ratio.

For winter conventional practice of electrical heater demands 5.5 kW as winter heating comes out to be 50 (Table D). So let us represent energy for winter $= P_3 = 5.5$ kW

$$\therefore \; COP_{heat\ pump} = 1 + COP_{refrigeration}$$

$$COP = 3$$

Present system takes $P_3' = \frac{5.5}{3} = 1.833$ kW

PI = 3 (approximate) was confirmed from the experimental result.

POWER AND COST ANALYSIS

EXISTING CONVENTIONAL SYSTEM

a. For Summer

4 months season:

Power $= 0.8 x P_1 x 122 x 10$ h/day

= 8540 kWh

Cost = Rs.0.8/kWhx6440 kWh

= Rs 5152

b. For Rainy

3 months season:

Power $= 0.8 x P_2 x 92 x 10$

= 6440 kWh

Cost = Rs. 0.8/kWhx6440 kWh

= Rs. 5152

c. For Winter

4 months season:

Power $= 0.8 x P_3 x 120 x 10$ hr/day

= 5280 kWh

Cost = Rs. 0.8/kWh x 5280

= Rs. 4224

HYBRIDIZED ALL THE YEAR ROUND SYSTEM

a. For Summer

4 months season:

Power $= 0.8 x P_1' x 122 x 10$ h/day

= 976 kWh

Cost = Rs.0.8/kWhx976

= Rs. 780.8

b. For Rainy

3 months season:

Power $= 0.8 x P_2' \; x 92 x 10$

= 4828.16 kWh

Cost = Rs. 0.8/kWhx4828.16

= Rs 3862.528

c. For Winter

4 months season:

Power $= 0.8 x P_3' \; x \; 120 \; x \; 10$

= 1759.68 kWh

Cost = Rs. 0.8/kWh x 1759.68

= Rs. 1407.75

Hence

For a 5 ton Plant	For a 5 Ton Plant
Total Power kWh/yr = 20260 kWh	Total Power kWh/yr = 7563.84
Cost = Rs. 16208	Cost = Rs. 6051 (Approx.)
Parallely, for a 10 Ton Capacity Yearly requirement of energy for a conventional system comes out to be 40520 kWh	Similarly, for a 10 Ton Capacity Annual Power requirement by the present modified system comes out to be 15127.68 kWh
Annual cost of energy = Rs. 32416	Energy cost/year = Rs. 12101 (approx
For a 100 Ton Plant	For a 100 Ton Plant,
Corresponding values for Power and cost are 405200 kWh and Rs. 3,24,160	Values of Power and cost for Modified system are 15127.8 kWh and Rs 1,21,020

Therefore, considering a life span of 20 years, the saving caused due to modified hybridized system for a 100 ton unit can be found out to be Rs. 4.06×10^6 or (40 lakhs).

Table D

Typical Cooling and Heating Load for 5, 10 and 100 Ton

Type of Load	5 Ton (17.5 kW) Summer (kW)	Winter	10 Ton (3.5) Summer	Winter	100 Ton (350 kW) Summer	Winter
$\dot{Q}_{structural}$	8	8	16	16	160	160
$\dot{Q}_{electrical}$	2	-2	4	-4	-40	-40
$\dot{Q}_{occupancy}$	3	-3	6	-6	-60	-60
$\dot{Q}_{Solar}$	2	-	4	-	40	-
$\dot{Q}_{ventilation}$	2.5	2.5	5	5	50	50
	17.5 kW	5.5 kW	35 kW	11 kW	350 kW	110 kW

U_{roof} = 0.87 kcal/h m^2C

U_{wall} = 0.94 kcal/h m^2C

As this type of modified system would be operating on water cooled condensing unit, one may take COP = 3, the values for the modified system (10 ton unit) are as given below:

Seasons	Power (kWh)	Cost (Rs)Yr.
Summer	1952	1561.6
Rainy	8586.66	6869.33
Winter	2640	2112
Total Power/Yr = 13178.66		Total = 10542.93

Thus for a 10 ton system corresponding yearly values of power and cost are calculated to be 13178.66 and Rs. 10542.93.

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